THE BAROMETRIC CONDENSER

Its Principles Simply Explained—Where The Energy Comes From—Effect of Air in the Tail Pipe —Descriptions of the Simpler Types

The atmosphere exerts a pressure at the sea level of, in round numbers, 15 pounds per square inch or:

$15 \times 144 = 2160$ pounds per square foot.

The pressure in the plane *AB*, Fig. 1, would therefore be 2160 pounds per square foot plus the pressure due to the hight *h* of water above that plane, and as it acts in all directions, it would tend to push the water up the pipe providing the atmospheric pressure had been totally removed from the interior *C*, until the weight of the column *CD* of water was equal to 2160 pounds for each square foot of its section, the columns *h* and *i* balancing each other.

Since one cubic foot weighs, in round numbers, 62 pounds, the column will have to be about:

$$2160 \div 62 = 34+ \text{ feet high,}$$

more or less according to the temperature of the water and the hight of the barometer.

Suppose we have an arrangement like Fig. 2, with exhaust steam coming into the chamber and injection water mingling with and condensing it and maintaining a vacuum of 26 inches or 13 pounds. This would support a column of water of only:

$$\frac{13 \times 144}{62} = 30 \text{ feet approximately,}$$

so that as the condensed steam and injection water fall upon the top of the column it would continually settle to the 30 feet which the difference between the atmospheric pressure and the absolute pressure in the condenser would support, and the condensate and the injection water would be continuously discharged from the vacuous space against the atmospheric pressure without the necessity of pumping.

In a jet condenser of the ordinary type, work has to be expended upon the mingled condensate and injection water to pump them out of the condenser against the pressure of the atmosphere. If the absolute pressure in the condenser is 2 pounds and the atmospheric pressure 15 pounds the water has to be pumped, as shown in the above calculation, against a head of about 30 feet to get it out against the unbalanced atmospheric pressure.

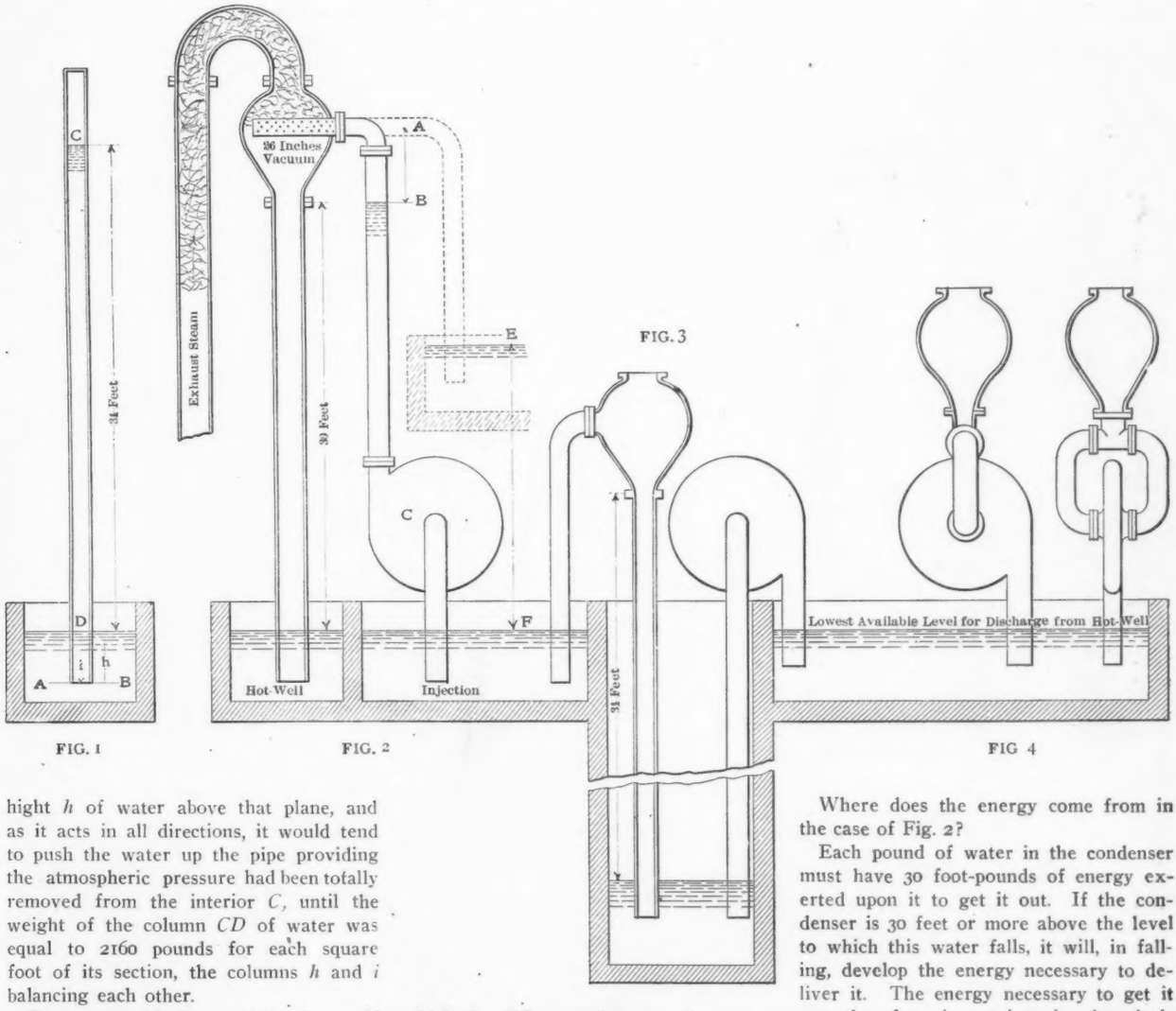

FIG. 1 FIG. 2 FIG. 3 FIG 4

Where does the energy come from in the case of Fig. 2?

Each pound of water in the condenser must have 30 foot-pounds of energy exerted upon it to get it out. If the condenser is 30 feet or more above the level to which this water falls, it will, in falling, develop the energy necessary to deliver it. The energy necessary to get it out, therefore, is put into it when it is elevated and its fall made possible, and comes from the source of the energy which elevates it.

So far as the steam is concerned, if the engine cylinder is above the condenser, as upon the large verticals, the energy has been put into the steam in lifting it to the cylinder. A five-thousand horse-power engine using 12 pounds of steam per hour per horse-power would take 60,000 pounds or 30 tons of steam per

hour, and this steam has to be lifted, either in the form of water or of steam, from the source of feed-water supply to the cylinder. If the cylinder is below the condenser the steam is pushed up the exhaust pipe by the excess of the terminal pressure over that in the condenser when the release occurs, and by the movement of the piston during the exhaust stroke, the pressure upon the piston being greater because of the column of steam which it is lifting.

In regard to the injection water, there

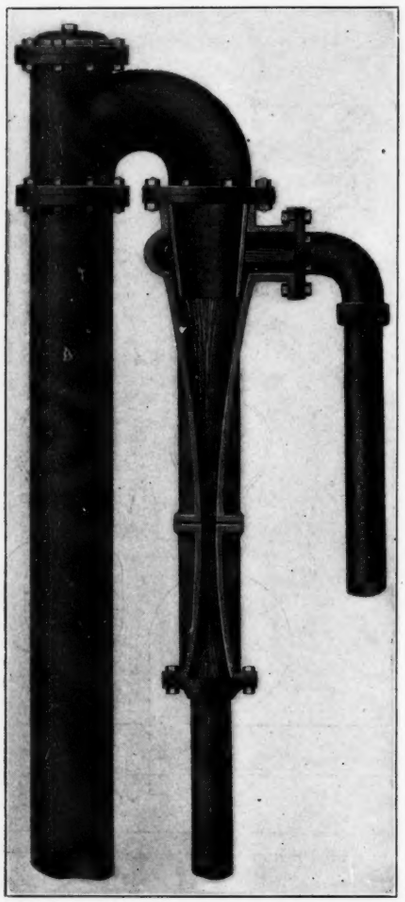

FIG. 5. THE BULKLEY CONDENSER

is no net consumption of energy (friction being neglected) in raising water to a given hight and letting it fall again, for in falling it develops as much energy as was expended in raising it. In Fig. 2 the pressure of the atmosphere would send the water to the level *B* and put into it the energy necessary to discharge itself against an equal pressure. The difference with the jet condenser is that it uses the atmospheric pressure to put the water in, but does not give it any chance to give back the work expended upon it in falling out.

Obviously the condenser cannot lift the water into itself from the level at which it discharges. An additional head *AB*, Fig. 2, must be overcome, and when the source of supply and place of discharge are upon the same level, a pump, as at *C*, must be used to overcome this additional head. If an elevated source of supply is available the condenser will draw its own water without any auxiliary, providing the head under which the water is supplied is such that the water in falling from *E* to *F* will overcome the head *AB* (always considering the abstract case without friction). The energy represented by the elevated position of the reservoir, the ability of the water to fall to the common lowest level, takes the place of the energy otherwise put into the pump *C*, and theoretically might be used to run the pump *C* and let the system proceed as represented in the full lines. If the head *EF* is greater than *AB*, the water would rush in too fast and artificial resistance must be put in in the form of a regulating valve; i.e., a part of the head *EF* must be sacrificed to overcome useless resistance.

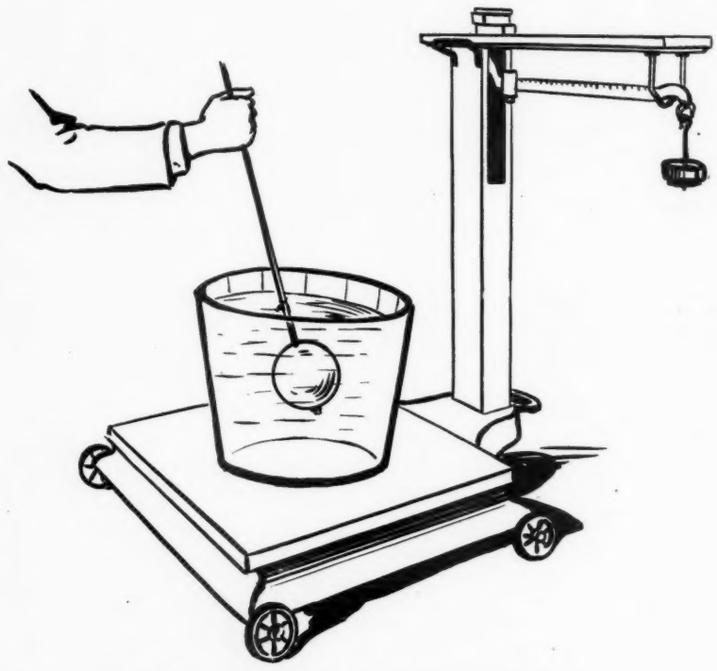

FIG. 6. TESTING THE EFFECT OF FLOATING OBJECTS

We might, with the two levels common to each other, lower the condenser until it would draw its water from the lower level as in Fig 3, but this would involve pumping out of the hot-well into the available discharge level instead of from that level to the condenser. It would be superfluous to use the long tail-pipe and the well as in Fig. 3, for since the columns of water in the descending and rising pipes balance each other they may as well be joined at any point in their length leading to a form of condenser which is now being developed by several makers. If the condenser head is lowered below the hight to which the atmospheric pressure will lift the injection water the helping column on the pump is decreased and its work increased. When we get down to Fig. 4 we have only to substitute the bucket pump for the centrifugal to have the ordinary jet condenser.

The condenser problem would be very simple if water alone were to be dealt with, but air is constantly coming into the system, not alone through leakage, but in the feed and injection waters. This air is non-condensible and expansive and would speedily destroy the vacuum if not removed.

A paper read before the Engineers' Society of Manchester, England, by W. E. Storey, credits James Watt with the invention of the barometric condenser and the dry-vacuum pump. Be that as it may, one Houpt was granted a patent for such a condenser in 1869 and there are said to be others prior to this date. That of Ransom was used considerably a half century ago. The tail-pipe was divided into a bundle of small tubes, at the summit of each of which a vortex was formed, such as is formed in a wash-bowl when the water is running out. This multiplicity of vortices was supposed to be necessary to draw out the air. The first to appear of those which have survived was that of Henry Bulkley, shown in Fig. 5. The injection water enters in the form of a hollow cone into the interior of which the steam is projected with the enormous velocity with which steam rushes into a vacuum, and the resulting mixture of condensed steam, air and injection water continues through the constricted neck into the tail-pipe with sufficient velocity to prevent any return of previously discharged air.

Anybody who has watched the discharge into the hot-well of a condenser must have noticed that a large volume of air was discharged into the water.

What is the effect of this mixture of air on the hight of the column in the tail-pipe?

A mixture of water and air is lighter than the same volume of solid water. It would appear, therefore, on first thought, that a given difference between the atmospheric and condenser pressures would support a greater hight of column of the air-laden mixture than of the solid water. This is not the fact. The pressure at the bottom of a column of water depends upon the hight of the column, no matter what may be the floating in the water. We will try to make this plain.

Balance an empty bucket upon a pair of scales with the pean. Put on the 100-pound weight and put water into the pail until the scales balance. The bucket will then contain 100 pounds of water. Note its depth. Now if a ball-float be submerged in it as shown in Fig. 6, one would have to push on it with a force equal to the difference between its own weight and the weight of an equal volume of water to keep it submerged. The scale then is supporting, and would show, if balanced over, an additional force equal to the weight of a volume of water equal to that of the float, made up of the weight of the float and the thrust upon it. The pressure upon the bottom of the pail then is greater than it was before by an amount equal to the weight of that volume of water. But the water is deeper and to restore it to its former level an amount of water equal to the volume of the float must be removed, which will not only restore the original depth but the original pressure per square inch at the bottom, notwithstanding there is less weight in the pail than before. Part of that pressure is due to the thrust of the hand; suppose the hand is removed, the float will rise; but during the time it is arising it will furnish a reaction for the buoyant effort and will keep the pressure upon the water just as it did when it was being held still. The pressure upon the bottom of the pail has been at all times, even when the float was rising freely, that due to the hight of the column of water above it, and this will be true in the tail-pipe no matter how much air is in it, so long as there is an unbroken, though crooked, column of water. When the bubble gets big enough to fill the pipe and break the column, that is another story.

If the bottom of the tail-pipe gets bared, gulps of air may be taken in, breaking up the column and lifting the water by the expansion of the air as in the Pohle pump. In order to provide against this action the hot-well should be so arranged that it cannot discharge below the foot of the tail-pipe, and when a pump is used to clear it the pump should draw from a compartment into which the hot-well overflows, and not from the compartment into which the tail-pipe delivers, so that it cannot pump that compartment below the end of the pipe.

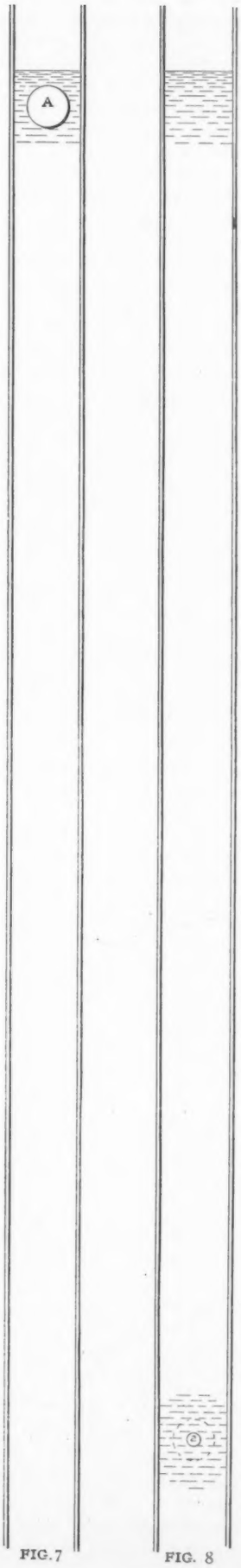

FIG. 7 FIG. 8

The air brought into the condenser under atmospheric pressure expands under the low pressure of the condenser to many times its original volume. The energy developed by this expansion is dissipated and is not available to recompress the air and expel it from the condenser.

Where does this energy come from?

Suppose a bubble to start with the size of *A*, in Fig. 7, up at the top of the col-

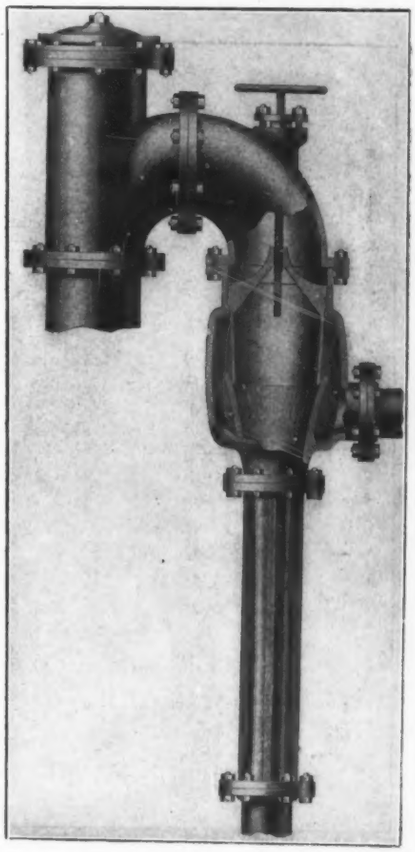

FIG. 9. THE BARAGWANATH CONDENSER

umn where the absolute pressure is 2 pounds. Suppose the column of water to descend faster than the bubble rises through it by buoyancy so that the bubble is carried down to the foot of the pipe where the absolute pressure is 15 pounds and is compressed to the volume represented at *a*, Fig. 8. It is evident, is it not, that more water would have fallen in the time of the bubble's descent than as though it had remained of its original size? A comparison of Figs. 7 and 8 will show that additional water equal to the volume of the compression (enough to fill the volume between the dotted outline of the original bubble and its volume after compression) would have fallen the full distance that the bubble did. The fall of

this additional amount of water is what furnishes the energy for the compression.

Going back to Fig. 5, we have the tail-pipe full of water with which a considerable quantity of air is mingled. The bubbles due to their buoyancy will start to rise in the column after they lose the downward velocity given them in the jet, and the column of water must be kept descending faster than they arise through it in order to carry them down to the hot-well. This is one thing which determines the amount of water which must be supplied.

Another thing is the amount of water required to condense the steam and cool it to the desirable temperature.

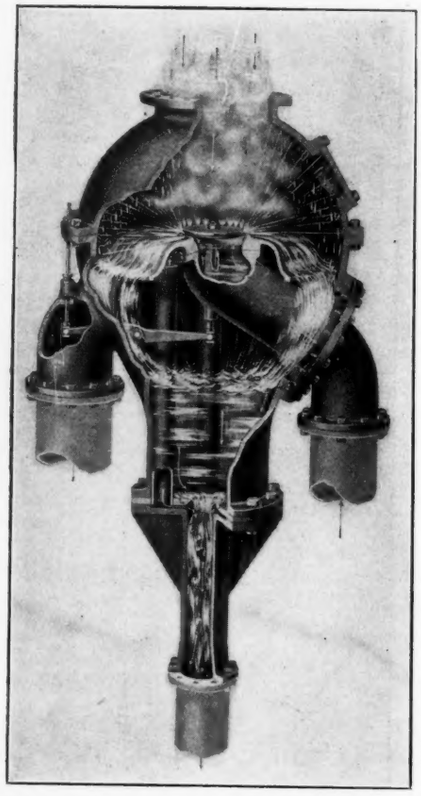

FIG. 10. THE TOMLINSON CONDENSER

There can be no absolute vacuum when water is present, for water gives off a vapor at every temperature. It is only at the atmospheric pressure that water boils at 212 degrees. Lower the pressure and it will boil at the lower temperature as follows:

Absolute Pressure.	Boiling Point.
14.7	212.
14	209.56
13	205.89
12	201 96
11	197 77
10	193.24
9	188.32
8	182.91
7	176.91
6	170.12
5	162.33
4	153.07
3	141.62
2	126.27
1	102.

If then we have a tube like Fig. 1 and start with an absolute vacuum in the space *A*, the water will send off vapor until it raises the pressure in that space to that corresponding with its temperature. If its temperature is 60 degrees, for instance, it will boil until it makes enough steam or vapor to raise the pressure in the space *A* to one-quarter of a pound absolute. It cannot boil under any higher pressure at 60 degrees so the pressure remains at this point. The higher the temperature of the water the higher the absolute pressure in the condenser, i.e., the worse the vacuum.

This is supposing there is nothing but water vapor present. What would happen if there were some air in the space besides the water vapor?

Every mixture of a gas and a vapor follows the following laws, called after Dalton, their discoverer:

1. *The tension and consequently the quantity of vapor which saturates a given space are the same for the same temperature whether the space contains a gas or is a vacuum.*

2. *The pressure of a mixture of a gas and a vapor is the same as the sum of the pressures which each would possess if it occupied the same space alone.*

The absolute pressure in the condenser then will be the sum of the vapor pressure due to the temperature and the air pressure. The less air there is present the nearer the absolute pressure will be to the steam or vapor pressure due to the temperature.

To get a given degree of vacuum, i.e., a desired absolute pressure in the condenser, one must then reduce the temperature of the mingled condensate and injection water to such a point that the sum of the steam pressure at that temperature and of the pressure of such air as may be expected to be present shall not exceed the desired absolute pressure.

The amount of injection water depends then upon:

1. The temperature of the injection.
2. The temperature of the hot-well.
3. The load.
4. The steam used per horse-power.
5. The initial pressure.
6. The amount of air.

The water will not flow out of the tail-pipe unless there is a greater pressure in that column than outside. Figs. 1 and 2 show a condition of equilibrium. In order to induce the flow the column must be somewhat higher than that which the atmosphere will support and the head inducing the flow will be that above the hight at which the column would balance the difference between the atmospheric and condenser pressures.

It is customary to make the leg 34 feet in hight, because, as was shown at the commencement of this article, this is the limit to which the water could be raised by a perfect vacuum, but as there can be no perfect vacuum in the presence of water the column cannot be raised to this hight by the atmosphere. A vacuum of 28 inches with a barometer of 30 would raise it to a hight of less than 32 feet. The difference is available to produce the downward flow, and to insure against damage if the column gets to surging. The greater the distance between the hight of the column supported by the atmosphere and the point of entry of the injection water, i.e., the distance *A B*, Fig. 2, the greater the work on the pump which lifts the injection water, and this difference must be great enough to allow a head on the column sufficient to produce a flow in the tail-pipe fast enough to carry off the air.

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In order to regulate the injection water and still preserve at light loads the velocity which is necessary to eject the air, William Baragwanath introduced the

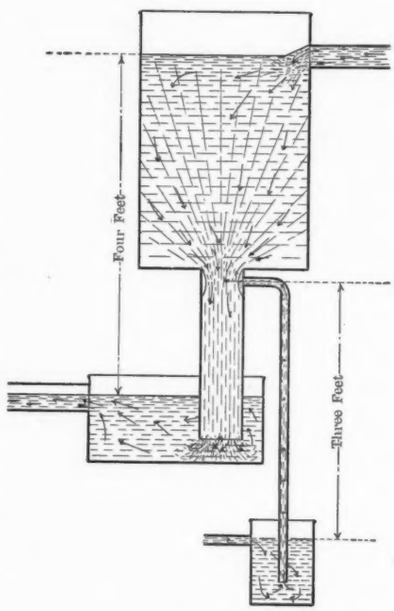

FIG. 11

regulating nozzle shown in Fig. 9. This regulates the thickness of the sheet of water entering the nozzle and permits a lesser amount to be introduced with undiminished velocity. But the amount of water required to condense the steam and reduce the temperature of the resulting water to the desired point may not be the limiting quantity. It may be necessary to put in more water than is necessary for this purpose at light loads in order to maintain the necessary velocity of flow in the tail-pipe to carry out the air. It is not practicable to make the diameter of the tail-pipe adjustable to meet this condition, but it is met in

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by the use of a double tail-pipe as shown in Fig. 10, which gives a sectional view of his Type *A* condenser for use without

a dry-vacuum pump. The injection water enters by the pipe at the right, and, controlled by the valve with the outside adjustment shown, is forced and broken over the serrated edges of the opening, sprayed into the steam and flows back over the umbrella, the contour of which is such as to throw the water outward against the shell.

The exhaust steam entering at the top mingles with the water, is condensed and falls with the water into the space beneath the umbrella and into the central tail-pipe. This pipe is proportioned so as to carry the water required for about two-thirds of the capacity for which the condenser is designed. When the flow exceeds this amount the water rises until it spills over into the auxiliary tail-pipe at the left. As this is not designed to carry off air, no particular velocity is necessary in it, and it simply stands at light loads with such a column of water in it as the atmosphere will balance.

A large amount of the air is carried by the water and the rush of the steam directly to the tail-pipe. There may be an accumulation, however, under the umbrella, and for the removal of this Mr. Tomlinson has applied a phenomenon described in several works upon hydraulics and illustrated diagrammatically in Fig. 11.

If water be allowed to flow through an aperture having sharp edges at the point of entry, the tendency of the water which enters from the sides to preserve its motion in a right line will produce a contraction of the stream just below the aperture, and the space surrounding the contraction will be vacuous. If a pipe be connected into it as shown it will be found that water can be lifted about three-quarters of the head inducing flow. If the distance between the levels of the large vessels in the diagram were four feet as indicated, the vacuous space would support a column of water three feet in hight.

The method of application of this action to the condenser is evident from the engraving. The entrance to the central tail-pipe is controlled by a plate with a thin edge as shown, producing the *vena contracta* and to the annular space between the two plates and surrounding the *contracta* the space beneath the umbrella is connected by vertical tubes. If two tail-pipes discharge into the same well, as is ordinarily the case, a baffle should be placed between them in order to prevent the return of air to the condenser through the auxiliary pipe.

(TO BE CONTINUED.)

There are today in use or in active contemplation in the iron and coal industries of Germany alone, four hundred large gas engines with a combined capacity of 420,000 horse-power.

SOME CONSIDERATIONS AFFECTING THE APPLICATION OF WASTE GASES FOR POWER PURPOSES

Utilizing Waste Gases from a Blast Furnace — Proportion of Power Gas Available from Coke Ovens — The Bearing of Natural Gas on the Problem — Best Disposition of Surplus Gases — Power from Gas-fired Steam Plant vs. Gas-Engine Plant — Effect of Gas Cleaning on Cost of Installation — Distribution of Power Outside the Works

BY F. E. JUNGE

While it is generally conceded that the various artificial gases which are available for the generation of power, such as blast-furnace, coke-oven and producer gases, must be cleaned in order to enable the prime mover to perform its service not only at a high degree of thermal excellence but also continuously and without breakdowns, it is not always obvious by which means the different gases can be cleaned most economically—that is, with as little expense for power, water and labor as possible; also what is the effect of gas cleaning on the efficiency of gas-fired boilers, gas engines, piping, etc. Further, very little knowledge has been propagated on such questions as: What is the best use that can be made of the gas inside the plant, and which is the most profitable way of utilizing the surplus quantity outside?

I have treated of the first part of the problem in detail elsewhere* and it is the object of this article to lay down such general data as are available on the latter application. It is obvious that with three different kinds of gases available, which show different characteristics, i.e., heat value, composition, temperature and impurities contained, not only relatively to each other, but each in itself at different times within comparatively short periods, the selection of the particular application to which each gas is best adapted is a matter of no little consideration and consequence.

BLAST-FURNACE GAS

Speaking more particularly of the utilization of blast-furnace gas in the iron and steel industry, which affords, indeed, the most complete exemplification of all the conditions which affect the operation of a gas power plant, it is known that for various well understood reasons blast-furnace works are preferably combined with steel-smelting plants and rolling mills in order to be able to carry out the entire series of converting and finishing processes which transform the original ore into marketable steel products, all under one ownership and with maximum industrial economy.

There are several distinct uses to which blast-furnace gas can be put in works of this magnitude, and the soundness of judgment exercised by the designer of the plant in the distribution of such uses will determine, on the one hand, the amount of additional solid fuel that is consumed in the plant in the form of coal or coke and that must be supplied by the works management at extra cost, and, on the other hand, the available surplus power that may be sold to advantage in the neighboring districts or cities, and which will yield remunerative returns in addition to the savings effected within the works.

The question whether the gas that is produced in the blast-furnace plant is better utilized for the production of heat by firing (besides the hot-blast stoves) open-hearth and other furnaces, or for the generation of motive power in driving blowing engines, rolling mills and central stations, can only be decided after a careful consideration of *all* factors which determine the commercial economy of a plant of this character. Thus we must primarily analyze the relation of the utilization coefficient of the first application, i.e., gas used for heating purposes, to that of the second, i.e., gas for producing power. In some instances, as with the heating of the hot-blast stoves and the driving of blowing engines, the values of the utilization coefficient and load factor are practically identical; but in others there may be considerable difference. So it will often be found that the utilization coefficient of gas for motive power cannot be estimated higher than from 60 to 75 per cent., while the corresponding item of gas for heating purposes may run as high as 85 and even 90 per cent. This difference is naturally founded on the intermittent working of the power plant with its auxiliaries, the various accessories, like pumps, hoists, fans, etc., showing together an extremely fluctuating load curve, while the various furnaces work almost all day and night continuously, consuming gas at a nearly constant rate.

The analysis must further embrace a careful comparison of the intrinsic values of the respective fuels which are displaced by the utilization of blast-furnace gas. Thus, if blast-furnace gas is used for raising steam, it displaces coal of inferior

*The Cleaning of Blast-furnace Gas, *The Iron Age*, September, 1906.

quality, while when it is used for firing furnaces it displaces gas coal of higher intrinsic value; and though it is difficult to estimate accurately the gain in favor of the latter utilization in exact figures, it must be conceded that in some cases such application will increase the working efficiency of the plant.

Moreover, there is to be determined whether the lower first cost of the heating appliances against those of a motive-power equipment is a decisive factor in favor of the former application, or whether the far superior efficiency of energy transformation in the latter method is of weightier commercial bearing.

GAS AVAILABLE FROM COKE OVENS

In some special cases, namely, when coal mines are located so near the works that they fall into the commercial distribution radius of the combined iron and steel-smelting plant, this part of the problem becomes even more complex, since of the total quantity of gas produced in coke ovens about 60 per cent. is used to heat the retorts, 10 per cent. to drive the various appliances of the coking plant—such as washers, pumps, etc.—while about 30 per cent. of the gas is available for outside purposes. Now, the relation of the respective intrinsic values, embracing heat contents, cost of generation, transmission and cleaning, of coke-oven versus blast-furnace gas, must be analyzed, since it may be found advisable to heat the steel furnaces with coke-oven gas of high calorific value instead of with the weak blast-furnace gas, thereby saving the cost of regeneration of the gas and having only regenerative ovens for the air supply, similar to what obtains when natural gas is employed for such purposes.

Assuming a consumption of only one ton of coke per ton of pig iron smelted, there is needed for a production of say 1200 tons of iron a day 1200 tons of coke. Figuring on an efficiency of transformation of 76 per cent., 1580 tons of coal are needed for making that coke. The total quantity of gas generated per ton of coking coal averages 28 cubic meters (988½ cubic feet), so that 442,400 cubic meters (15,623,000 cubic feet) of coke-oven gas are produced within 24 hours. This, of course, is only an assumed amount, since in modern by-product ovens the quantity of gas produced depends on the quality of the coal coked, on its moisture contents, and on the type of oven, and varies considerably in composition during one coking period. In the latest regenerative ovens of the Otto type up to 140 cubic meters (4840 cubic feet) of gas per ton of coal coked are attained, the gas consisting chiefly of CH_4 and H, and having a calorific value of about 4000 calories per cubic meter (448 B.t.u. per cubic foot). About 60 per cent. of this is used for heating the retorts, leaving 40 per cent. for other purposes. This gas has a calorific vaue of 500 B.t.u. per cubic foot and some 25 cubic feet per hour of it, when burned in a gas engine, will develop one horse-power. The total available energy of such a plant would therefore be 10,500 horse-power. Of this amount about 10 per cent. is used for driving plant auxiliaries, leaving 9500 horse-power available for sale.

If the coke ovens are located near the steel works the gas may be used in the works for heating steel furnaces, as stated above. In a plant of the above capacity this would mean 1,836,360 cubic feet of coke-oven gas displacing 43 tons of good coal and absorbing one-third of the total surplus quantity of gas available (5,650,000 cubic feet). Without deducting the amount for the above application, there is for every ton of coal transformed to coke in 24 hours 6 horse-power available for other uses.

PRODUCER GAS

Nor is this question nearly settled with the foregoing considerations, since in combined plants of this magnitude there are still other resources available, such as inferior grades of coal from the mines, culm piles, etc., which, though hitherto wasted, can now by the application of up-to-date methods be fully utilized for the economic generation of heat and power gas, in addition to what is gained as a by-product from the blast furnace and coke oven. Such practice is now finding universal adoption in Germany, since several years of experience with the Jahns type of ring producer and other systems have proven its practical merits beyond discussion. A plant of this kind has done active service in the von der Heydt coal mines since April, 1903, which is a sufficient time for drawing definite conclusions as to results. The fuel used is slack, residue and refuse which drop from the coal conveyers and tipplers; also culm banks, which were formerly wasted. It only contains 20 per cent. of coal and is now fed directly to the producers. In this way 2100 tons of waste material are gasified, giving a total of 14,000,000 B.t.u., or 3245 B.t.u. per pound. The cost of 1000 B.t.u. is 0.005 cent.

Of the heat developed, 13,650,000 B.t.u. are used to generate 3500 tons of steam. One ton of steam from gas-fired boilers costs, therefore, 20 cents for fuel, as against 44 cents from coal-fired boilers, as a certain quantity of steam coal has to be supplied in addition to the waste in order to meet the demand. Part of the gas is used in gas engines for the generation of electric power. The cost of the gas per brake horse-power hour, assuming a consumption of 9750 B.t.u., comes out as 0.05 cent. The steam cost per brake horse-power-hour in steam engines is found to be 0.51 cent when steam is raised in coal-fired boilers, and 0.24 cent when it is raised in gas-fired boilers. Figuring on an average consumption of 10,000 B.t.u. per hour per brake horse-power in gas engines, and deducting losses through natural deterioration of the fuel, it can be taken that one ton of culm generates in modern producers from 20 to 25 horse-power for 24 hours.

It is by combinations of such character and magnitude that the iron industry affords the most striking and comprehensive field for the application of gas power from a variety of sources and for a multitude of purposes. Indeed, leaving aside natural gas, which, owing to its territorial and quantitative limitations, cannot claim consideration in this discussion, we find all the principal sources of commercial gas generation, namely, the blast furnace, the coke oven, and the producer, as well as all forms of transformation, namely, heat, light, electric energy and mechanical power, and their modes of distribution, represented and combined in this one field. Therefore, economic considerations, commercial questions, and technical research on the production and utilization of gas may always be based on the iron industry as the most fitting subject for such studies.

RELATION OF NATURAL GAS TO WASTE GASES

It was said above that natural gas cannot claim consideration as a fuel for large-scale operations in the iron and coal industries. This, of course, refers only to future activities. When natural gas was first discovered and brought into practical use there seemed to be the general idea that the supply was inexhaustible, and it was sold at low rates and usually without measurement. This method encouraged waste in the consumption of natural gas and was only abandoned after the large companies had obtained control of the business. But the gas which was wasted in the early period of production cannot now be regained by recourse to economic methods of distribution and consumption.

As far as the present production of natural gas is concerned, the increased value in 1905 ($41,562,855 compared to $38,496,760 in 1904) is recorded by the U. S. Geological Survey to have resulted from a general advance in price rather than from any increase in yield. As a matter of fact the great gas fields of Indiana and elsewhere have shown a steady decline since 1902, and the value last year was considerably less than one-half of the maximum production. And even conceding that in several States large and prolific gas fields are being opened up, this would not be of much consequence to the iron and coal industries as consumers of gas power, who must have a definite guarantee that the supply of fuel on which the constancy of production is founded will be upheld for an indefinite time, in unchanging quantities, and within the commercial distribution sphere of their works.

Where natural gas is available as a by-product of the property owned by some manufacturing concern, or where it can be had in the immediate vicinity of a plant, then it goes without saying that it

will be used, provided that it can be bought at a reasonable price. It may even be pumped over long distances provided that this operation does not make it non-competitive with the available blast-furnace and coke-oven gases. Owing to its high heat value, which ranges from 900 to 1000 B.t.u., and to its great heat density, it is the ideal fuel for transportation, being greatly superior to coke-oven and blast-furnace gases, especially the latter, which has a thermal value of only 90 to 100 B.t.u. per cubic foot.

A further advantage is that no additional expenditures for cleaning have to be charged against natural gas, when burnt in gas engines instead of under boilers, while with blast-furnace gas a part of the cost of cleaning and with coke-oven gas the total amount must be charged against it in addition to the price at which the works management appraises the different "waste" gases. For, when heating boilers, coke-oven gas need not be specially cleaned and yet will give better results than the weak blast-furnace gas, which must be purified and freed from dust in order that the results attained may be similar to those with coal-fired boilers.

Thus the merits of gas power versus steam power, which will be discussed presently, are less pronounced in collieries where coke-oven gases are available. This is partly due to the fact that the cost of power generation represents not nearly so large an item as it does in iron and steel works, since, unless distribution to neighboring districts is provided or very unfavorable conditions prevail, not more than four to five per cent. of the total quantity of coal produced (or its equivalent in form of waste heat or waste gases) is required in the mines, provided they are equipped with modern economic power plants. It is also partly because in contradistinction to steam drive, the coke-oven gas engine must be charged with the total cost of gas cleaning, in addition to the price charged for the gas as produced. The latter valuation depends entirely upon local conditions. It is sometimes based on a rate corresponding either to a certain weight of coal of thermal equivalence, the price of which in turn depends on whether it is purchased from other or from one's own mines, or to the amount of steam that can be generated by a certain measure of both fuels, or to that of some other standard of comparison.

In any case the valuation of what is called waste in industrial pursuits is nowadays a matter of no mean importance and is not determined and dependent on purely theoretical and technical considerations, but on practical economic questions which have a decided bearing on the remunerative returns of the capital locked up in the different branches of a large industrial concern.

Summing up, it was said that natural gas is an ideal fuel for heating furnaces, raising steam under boilers, and serving to operate gas engines. The fact that over 100,000 horse-power are generated in gas engines running on natural gas in this country is a better proof of its adaptability to that use than any other argument. But against all these advantages there stands the other fact that the production of natural gas is on the decline, while the demands for gas power are increasing daily. Therefore it is safer to base our claims for future activities on coal as the energy-supplying fuel, since it is certain that the annual production in this country of nearly 400,000,000 tons can be kept up for hundreds of years to come.

UTILIZATION OF AVAILABLE POWER GAS

Now coming back to our discussion of the principal conditions which determine the commercial and technical distribution of these various gases within the works, a new problem presents itself. After having decided to apply a certain quantity of a certain gas to the production of motive power, the other no less important question arises whether it is more advantageous to use the gas under boilers for steam raising and to employ steam turbines in the central station, or whether it is more economical to burn the gas directly in gas engines, the points in favor of the first equipment being lower first cost, smaller floor space, less expenditure for up-keep, and no difficulty to secure skilled labor, while the advocates of gas power advance arguments of no less weight, namely, elimination of the wasteful and costly boiler equipment with its danger of explosion, smoke nuisance, etc., and reduction of operating cost to one-third of the cost of steam drive.

Since earnest and laudable efforts have recently been made on the part of gas-engine manufacturers to reduce the first cost price per unit of output to the level of steam-engine costs, and since the only difference between the two forms of application consists in the employment with gas power of a cleaning plant as substitute for the boiler equipment, the whole controversy resolves itself into the very simple requirement to provide for a gas-cleaning plant, which is superior in economy, as regards first cost, floor space, water consumption, and skilled labor, to a steam-boiler plant of the same capacity. But even this is no longer a correct argument, since it has been found that in order to get the highest plant efficiency with steam the gas for boiler heating must be brought to almost the same degree of purity as when burnt in gas engines. This will be discussed later. All other objections are insignificant compared to the simple fact that by the direct utilization of the gas in gas engines, power can be produced at one-third of the cost that is on record for any other form of power generation.

When iron works and coal mines are located in the neighborhood of other industrial centers, communities or cities, and provided they have a sufficient amount of salable surplus power available, the works management is confronted by another problem, namely, to decide which system shall be adopted for the supply of these outside markets. They can have the gas-cleaning or by-product recovery plant put up near the furnace and at their own expense, and deliver pure gas to the power station or works to be supplied, or the owners of the iron works and coal mines can put up the cleaning or recovery plant and a complete electric power station, which may, of course, be combined with the central station of the works, and sell the electric energy to the supply company or works at so much per unit. Which of the two methods is the more advisable to adopt depends entirely on local conditions.

The foregoing reference to the power question was made in order to show, by a careful analysis of the actual conditions prevailing in the iron and coal industries, what significance the factor of gas cleaning possesses in the general problem of securing maximum industrial economy from the utilization of blast-furnace and other available gas. With certain limitations the same line of thought commends itself also for the design and operation of collieries. But, leaving a detail discussion of the latter application for later consideration, I shall first attack the subject from a different point of view, and one that will embrace the enumeration of practical advantages gained by the adoption of efficient methods of gas cleaning in such plants, as well as the cost of gaining them.

CLEANING IS REQUIRED FOR HEATING AS WELL AS FOR POWER GAS

It was stated in an earlier part of this article that there was until a short while ago very little actual experience available on the matter of gas cleaning, and that it was held by eminent authorities that if the larger part of the gritty dust contained in the blast-furnace gas were removed in the dry-dust catcher the remainder would not prove harmful to the stoves, boilers and engines to which it was supplied. It was also maintained that the gas should never be washed for boiler heating, as any tarry products it might contain would enhance its heating power by increasing the luminosity of the flame. Furthermore, there was the seemingly weighty argument submitted that the cost of cleaning the gas, together with the increased plant floor space, were apt to annul the advantages gained from the superior heating properties of the cleaned gas.

Consequently, in the first attempts to utilize the waste gases from blast-furnace plants the hot gas was delivered directly to hot-blast stoves, steam boilers and furnaces laden with dust and at a temperature of from 140 to 160 degrees Centigrade. Needless to say that with such

practice it was necessary periodically to shut down the boiler plant for cleaning the settings, besides the cleaning that was ordinarily done as a part of the daily routine of the works, and that the frequent cooling of the boilers subjected them to heavy strains, which greatly impaired their efficiency and necessitated frequent repairs. Furthermore, the heating surface of the boilers would gradually decrease on account of the dust that settled down at a cumulative rate, thereby requiring a constantly increasing quantity of gas for generating the same amount of power.

The lower heating value of uncleaned blast-furnace gas per unit of volume, and its inferior combustion efficiency when containing considerable quantities of fine dust, would anyhow necessitate a larger grate area of boilers, the difference compared to the employment of clean gas running as high as 10 per cent. At the Cockerill works in Seraing, Belgium, it was found that after cleaning a boiler and putting it into commission again it required with dirty gas three hours' time to get up the steam pressure, while by using clean gas this time could be reduced to one and one-half hours.

A. Gouvy records a case established by actual measurements where, with a freshly cleaned fire-tube boiler, the consumption of blast-furnace gas amounted to 1925 cubic feet per pound of water evaporated, while after a fortnight's operation the consumption increased to 3529 cubic feet, or almost double the amount. This increase is explained by the dust accumulating in the fire tubes and forming a thick coating over the heating surfaces of the boiler. It was also shown by experiment that the cleaned gas effected a larger evaporation per unit of heating surface with less consumption.

In hot-blast stoves of the Cowper type the cost of up-keep is not a very important item of expense, since even at present their internal structure can be maintained in good shape for three years and longer. What the cleaning of gas does in this instance is that the heat-radiating capacity of the fire brick is greatly increased, since it is no longer covered with dust and slack, so that with the same quantity of gas higher blast temperatures can be attained and a great saving in coke consumption effected.

Gouvy's experiments prove that for the higher temperature limits of from 650 to 900 degrees Centigrade an increase of the blast temperature by 100 degrees will save from 110 to 165 pounds of coke per ton of pig iron produced. For the lower limits this saving is even higher and runs up as high as 220 pounds per ton of pig iron smelted. Assuming the price of coke to be $3.60, the cleaning of the gas would effect a reduction of the cost of production of pig iron by at least 18 cents per ton, if the temperature of the blast were increased by 100 degrees Centigrade. A blast-furnace plant working with very high blast temperatures, such as from 850 to 900 degrees, will, of course, be unable to effect a saving in the above sense, but the cleaning of gas enables one to use less gas in the hot-blast stoves and to employ the resulting surplus for heating boilers, which again results in a reduction of the coal bill. At Dommeldingen, Germany, the employment of cleaned gas for heating Cowper stoves has effected a reduction of coke consumption, in consequence of the higher temperatures attained, representing an annual saving of $8600 for one 100-ton oven.

All these deficiencies of operation, and more especially the considerable amount of manual labor that had to be expended for removing the dust, have served to convince the designers of gas-power plants that it is more economical to clean that portion of the blast-furnace gas which is used for raising steam under boilers, but then again it was maintained, for reasons which cannot very well be defined, that it was unnecessary to extend purification to that other part of the gas which is used for heating the blast. Yet it is clear that the same line of thought that leads to the cleaning of the boiler gas must also bring about similar advantages when extended to the gas heating the blast stoves.

Thus one of the stoves which is now installed to serve as a spare unit for reserve, in order that the capacity of the furnace may be maintained during the period of cleaning of the different stoves, may be done away with entirely, thereby saving considerably in the first cost of the installation. Smaller heating surface, superior combustion efficiency, and higher temperatures in the blast stoves, besides the saving in labor for removing the dust, are some of the other advantages gained. Depending on the design and capacity of the stoves, the percentage of moisture contained in the air, the intensity of radiation, the quality of coke charged with the ore and the composition of the latter, and on the degree of purity of the gas, it is possible to reduce the quantity of gas supplied for heating the blast to from 25 to 18 per cent.

VARYING QUALITY OF BLAST-FURNACE AND COKE-OVEN GASES

One more point, which comes up when studying the physical and chemical properties of blast-furnace gas as an energy-transforming medium, is deserving of consideration, namely, the varying quality of the gas at different stages of the generation process. While the average thermal value of blast-furnace gas lies between the limits of from 100 to 106 B.t.u. per cubic foot, and sometimes even reaches higher values, it often drops down to 90 or 85 B.t.u. per cubic foot in the course of one day's operation. In coke-oven practice the composition of the gas changes even within wider limits. Owing to the working process of the standard type of four-stroke-cycle gas engine, the output of which is rigidly limited by the cylinder suction capacity, the power plant, even when equipped with ample gas storage, is unable to sustain peak loads for any length of time while the energy of its working medium is thus widely fluctuating.

Though the automatic gas-supply system which has been introduced by the writer in the design of such plants eliminates this trouble of inflexibility of engines by providing between the source of gas generation and the prime mover an elastic member, preferably a fan running at variable speeds, with its output automatically regulated from the governor of the engine according to the momentary requirements, thereby securing flexibility and overload capacity similar to steam drive, it is yet desirable that regularity and uniformity of the conditions which affect the working of the furnace directly be likewise maintained. Therefore, if the varying composition of the blast-furnace gas is uncontrollable, the maintenance of a constant degree of purity of the gas used for heating the blast, as well as the permanent efficiency of the heating surface of the blast stoves, is imperative, as these features increase the working efficiency of the plant. For it must be remembered that all these various niceties of operation help to reduce the amount of manual labor that has to be expended, as well as the quantity of gas required for generating heat and power to carry out the long series of reducing, converting, and finishing processes, thereby eventually decreasing the coke consumption per ton of pig iron smelted, and hence also the total cost of production.

EFFECT OF GAS CLEANING ON COST OF INSTALLATION

Another factor which has hitherto not been fully appreciated is the influence of the reduction of temperature and volume of the gas, which is effected by the cleaning process, on the capacity and first cost of the installation and also on the cleaning process itself. Consideration of the temperature-pressure relations of a gas, or better the ratio of the increase of density to the decrease in temperature, will show that the gain effected by a reduction of temperature of 100 degrees Centigrade is represented by a contraction of the gas to one-third of its original volume. Now it is obvious that the capacity of an apparatus wherein the gas is transformed by combustion into heat or power, or in which it is washed or mixed or moved, will be increased in a similar ratio while the efficiency of such processes also becomes better. Therefore, to secure a reasonable degree of engine capacity with blast-furnace gas, which is by far the leanest of all commercial gases, it is necessary to reduce its temperature to some 25 or 30 degrees Centigrade. This is, of course, also dictated by other conditions, such as danger of premature ignition, etc.

The range of temperature reduction in blast-furnace work is from about 150 degrees Centigrade to about 25 degrees, that is, a reduction of about 125 degrees Centigrade, and it can easily be computed what shrinkage is effected by such cooling; also what the effect of the contraction is on the size and dimensions of the conduits, pipes, and channels used for conveying the gas. The latter point is further emphasized by the fact that the gas-cleaning plant delivers the gas to boilers, engines, and stoves under a pressure of from two to four inches of water, depending on the kind of fan or blower employed, thereby further reducing the bulk of the distributing means and the first cost of the installation.

The foregoing enumeration of reasons may suffice to prove the necessity for subjecting the whole quantity of gas generated in blast furnaces, coke ovens, and producers to thorough purification before utilizing it for heating blast stoves or furnaces, for raising steam under boilers, or for generating mechanical or electrical power in the central station.

COST OF GAS-CLEANING EQUIPMENT

In order to give some idea of the additional expense introduced by the instalment of a modern cleaning plant of the high-speed centrifugal type, the following data from European practice are presented, all items being based on 1000 cubic feet of gas. The initial capital outlay is $26 for heating gas and $39 for power gas, not including pipe connections, clearing ponds, water pumps, gas holders and similar accessories. The power for moving the gas and pumping the water is 0.15 horse-power for heating gas and 0.27 horse-power for power gas. The consumption of cooling water per hour is 31.2 gallons for heating gas and 46.4 gallons for power gas, the temperature of the water varying from 5 to 35 degrees Centigrade, according to the season. The cost of operation for plants in use 365 days in the year and 24 hours daily, figuring on the horse-power as costing 0.25 cent, and ignoring the value of the blast-furnace gas and amortization, but including regular attendance, oil consumption, repairs and upkeep, in German practice comes out 0.04 cent for heating gas and 0.072 cent for power gas. With the above a degree of purity of one grain of dust per 1000 cubic feet of gas for use in gas engines can be attained.

THE OUTSIDE DISTRIBUTION OF POWER

A reference was made in the above to the distribution of surplus power from iron and steel works and collieries to neighboring districts. The most desirable means of long-distance transmission is, of course, electric current, though serious propositions have recently been made to pump gas from the coalfields over long distances to cities, to be there used for heat, light and power purposes in competition with the existing gas companies. Of the available artificial gases coke-oven gas is superior to blast-furnace gas for transmission, owing to its greater heat density. In oil regions the employment of oil gas is going to be a factor of competition of no mean importance.

The surplus power which remains available after deducting all requirements within the works may be utilized in various ways. The most feasible and the one most often used is to deliver light and power to communities or cities located in the immediate vicinity of the plant. Where these natural outlets are not available, it is becoming more and more customary to establish a system of power exchange between such works as are located within a certain commercial distribution sphere relatively to each other, the object being to save in initial and operating cost of plants, to balance the stability of output by a corresponding provision for consumption; in other words, to obtain a constant high-load factor for the combination of works, and last, but not least, to provide for a common and ample source of energy in cases of emergency.

Thus small coal mines which produce good coal for coking purposes, and have sufficient coking capacity, will install a coke-oven gas power plant which, besides furnishing energy to these mines, distributes electric power in form of high-tension electric current to neighboring mines at a profit, or they will make an agreement with some central electric station to deliver a certain amount of energy at a certain figure all the year round to the said station, whence it will be sold and distributed to consumers at a profit. Since the central station holds similar contracts with a number of individual contributors, and since the agreement provides that in case of a break-down any contributor may become a consumer, that is, may take energy for emergency uses within his works from the line, it is evident that this arrangement offers an ideal means of making and selling energy under economic conditions profitable to all alike over wide territories, provided that the respective concerns can persuade themselves that a combination of such mutually beneficial character does not preclude competition.

The saving effected through these combinations is due, on the one hand, to the elimination of special power generating units and costly reserves; on the other to the maintenance of a constant high load factor. The importance of the latter item will be appreciated when it is considered that in some gas power plants an increase of the load factor from 25 to 50 per cent. will halve the total cost of power.

For the practical realization of reciprocal policy we must again turn to Germany, where, owing to the close concentration of industrial centers, most headway has been made in the centralization of power. In a recent paper on the production and utilization of power in metallurgical and mining pursuits Dr. Hoffman, of Bochum, discusses quite a number of such combinations, of which the one covering part of the territory of Rhenish Westphalia is the most interesting, because it is the largest of its kind up to this date.

The price paid by the Rhenish-Westphalian Electric Central Station to the various iron-smelting plants and collieries for their surplus energy is 0.7 cent per kilowatt-hour. Owing to its own large power plants, aggregating about 55,000 horse-power, and to the fact that it can save the reserves, it is possible to sell energy at a very low rate.

For instance, to large consumers, such as factories, rolling mills, collieries, etc., the charge is 1.4 cents per kilowatt-hour, and for driving motors which run at a constant high load factor, such as fans for mine ventilation, the charge is as low as one cent. This central station has also made contracts with several communities and cities, and has taken active part in the operation of smaller central stations and of electric railroads running through the commercial distribution sphere of their works. The tariff rates are based on a sliding scale, beginning at 7.6 per kilowatt-hour for light, and 3.5 cents for power, and decreasing in proportion to the consumption to 3.5 and 1.4 cents respectively.

Similarly those collieries which transform a considerable portion of their coal output to coke, are selling their surplus energy to neighboring districts. The largest German coal mine, "Rheinpreussen," will shortly produce 3,000,000 tons per annum, of which one-third is to be coked. With this amount at least 17,000 horse-power is generated in gas engines, while only 10,000 is used in the collieries. The surplus energy is sent in form of electric current at 10,000 to 20,000 volts, to the city of Krefeld and its new Rhein harbor, where it is transformed and sold at 1.9 and 1.7 cents per kilowatt-hour respectively.

Another and very promising way of disposing of the surplus energy is to utilize it for driving electric railways through the commercial distribution sphere. This proposition is interesting, both commercially and technically, and will be made the subject of a separate article. The employment of cheap gas power is destined to become a promoting factor of great weight in the electrification, also, of long distance railroads.

If one keeps in mind the facts that the application of gas power in the iron and coal industries from the blast furnace and coke oven, disregarding entirely the utilization of culm, will generate in the neighborhood of four and a half million horse-power the year around, and, in addition to what is consumed within the works will liberate an enormous amount of surplus energy which may be supplied to neighboring districts in form of heat or light or power, and that the actual saving thereby effected in

the iron industry amounts under favorable conditions to one dollar per ton of pig produced and to three or four dollars per ton of finished goods turned out, then considerations of the character developed in this article will doubtless be given more attention than they would without referring to the extreme economic importance of the problem.

Catechism of Electricity—XVIII

ROTARY CONVERTERS

366. *Is it possible to change alternating currents into direct currents?*

Yes; one of the most practical ways of doing this is by means of a "rotary converter."

367. *What is a rotary converter?*

A direct-current generator equipped with collector rings which are joined by means of conductors to certain symmetrically spaced parts of the armature windings. Such a machine does practically the same work that would be done by a direct-current generator direct-driven by an alternating-current motor.

368. *Are all rotary converters of the same type?*

No; there are two kinds of rotary converters: the direct-current-field type and the induction-field type. As the latter must be more carefully designed for the the circuits on which they are to be run and are considerably more sensitive to disturbances than the former type, the majority of rotary converters in commercial use are the direct-current-field type. The field-magnet of this type is exactly like that of a direct-current dynamo; hence the name.

369. *Show the armature connections of a rotary converter of the direct-current-field type.*

Fig. 81 shows the armature connections of a single-phase, six-pole rotary converter of this type. The part *A* shows the alternating-current or motor side of the machine, and the part *B* shows the direct-current or generator side. These are opposite ends of the same armature and of the same set of magnet poles. Both the armature and the magnet poles are wound as they would be for any direct-current, six-pole machine.

The connections on the alternating-current side *A* will perhaps be best understood by referring first to a simple direct-current, two-pole generator. To convert this into a single-phase alternating-current motor, it is necessary merely to connect two diametrically opposite points of the armature winding to two separate collector rings. In Fig. 1 there are six field-magnet poles instead of two as in the case just considered, so that instead of joining two equidistant points in the armature winding to the collector rings, 3×2 or 6 equidistant points around the armature are thus alternately connected to the two collector rings *c* and *d*.

370. *Explain the operation of the converter armature just described.*

With a single-phase alternating electromotive force applied to the collector rings *c* and *d* by means of the brushes pressing on them, and with the field-magnet excited, the armature, when once started will continue to rotate, and will run as an alternating-current motor. Part of the alternating current fed into the armature for running it is led to the commutator *o* on the direct-current side *B*, and is converted into direct current in the usual way. This direct currrent passes through the brushes pressing on the commutator and thence to the outside circuit *m h*. By reason of the rotation of the armature windings in the magnetic field, alternating electromotive forces are developed in them; these electromotive forces, however, oppose those applied at the alternating-current end. The current in the armature windings is due to the difference between the alternating electromotive forces and the direct-current electromotive forces; it is therefore comparatively small and produces but little heat in these conductors.

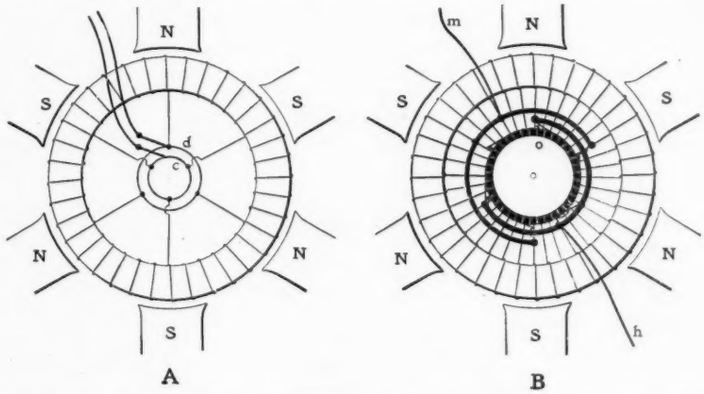


FIG. 81. ARMATURE CONNECTIONS OF A SINGLE-PHASE ROTARY CONVERTER

371. *Are single-phase rotary converters generally used?*

They are not, on account of their inability to start themselves as alternating-current motors. Two-phase and three-phase rotary converters are more satisfactory in this respect. They operate on the same principles as the single-phase converter, but two-phase converters have four collector rings connected so as to give two circuits through the armature winding under each pair of poles. The rings are connected to four equidistant points in each section of the armature winding covering the space spanned by one pair of poles. Three-phase converters have three collector rings connected so as to give three circuits through the armature winding under each pair of poles. The rings are connected to three equidistant points in each section of the armature winding covering the space occupied by one pair of poles.

372. *What other advantages have polyphase converters?*

The average path of the current through the armature from the collector rings to the brushes is short, so that the armature resistance loss and the armature reaction are small, and the efficiency high. On account of the low armature loss in comparison with that in direct-current generators of equal output, the load on a two-phase converter can be increased about 60 per cent., and the load on a three-phase converter can be increased about 30 per cent., above their respective outputs as direct-current machines.

373. *What relation exists between the alternating voltage applied to the motor side of a rotary converter and the direct-current voltage available at the generator side?*

In either a single-phase or a two-phase converter the applied alternating voltage is approximately 0.71 of the voltage on the direct-current side. In a three-phase converter the applied alternating voltage is approximately 0.61 of the direct-current voltage.

374. *To obtain direct current at 550 volts from a two-phase rotary converter, what alternating voltage must be applied?*

According to the preceding answer: $0.71 \times 550 = 390.5$ volts must be applied.

375. *What direct-current pressure would be developed in a three-phase converter supplied from a 500-volt alternating-current circuit?*

$500 \div 0.61 = 819.7$ volts would be developed.

376. *What effect has the frequency of the alternating-current circuit upon the operation of rotary converters?*

High frequency supply entails high peripheral speeds and narrow commutator bars and these introduce serious difficulties in the mechanical construction. Frequencies of 40 cycles, or better still, 25 cycles per second, give the most desirable results.

377. *How are rotary converters usually started?*

By applying alternating-current directly

to the collector rings, so that the converter starts as an alternating-current motor; by means of a separate alternating-current motor connected to the converter shaft; or by applying direct current to the commutator so that the converter starts as a direct-current shunt-wound motor.

378. *Describe in detail the starting of a rotary converter by applying alternating current to the collector rings.*

This method is applicable with polyphase converters but not with single-phase converters. The field winding is open-circuited at several points by means of a suitable switch, in order to limit the voltage induced in the field winding by the rotating magnetic field developed by the alternating current in the armature winding. The direct-current side of the machine is disconnected from its circuit during the starting process, as the current produced is alternating until synchronism is reached. The moment of synchronism can be judged by means of incandescent lamps connected across the leads from the brushes. When synchronism is reached, the lamps will burn

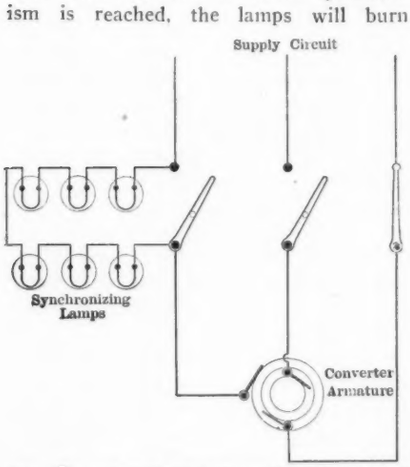


FIG. 82. SYNCHRONIZING CONNECTIONS OF A CONVERTER, FOR STARTING AS A DIRECT-CURRENT MOTOR

steadily. The field circuit may then be closed, the direct-current side connected to its circuit, and the load thrown on.

379. *Describe the starting of a rotary converter by means of a separate alternating-current motor connected to the converter shaft.*

The motor used for this purpose is of the "induction" type and is designed with a smaller number of poles than the converter, so that its normal speed will be higher than that of the converter and it will therefore be able to bring the converter up to synchronous speed. The converter armature is entirely disconnected from both the alternating-current and direct-current circuits, its field is excited and the induction motor is started up in the ordinary way. When the converter has been brought up to synchronism with the alternating-current circuit the switches in the leads to that circuit are closed. The induction motor is cut out of circuit, leaving the converter running as a synchronous motor. Then the direct-current switches are closed.

380. *Describe the starting of a rotary converter by applying direct current to the commutator.*

The converter is started as a shunt-wound motor. If the field-magnet is compound wound, the series-field coils must be short-circuited or left out of a circuit so that variations in the armature current will not affect the speed by varying the strength of the field. A strong field is given the converter in starting, and the current through the armature is gradually increased by means of a starting rheostat in series with the direct-current side of the converter. The speed of the converter is increased by means of the rheostat in the shunt-field circuit until the converter is in synchronism with the alternating-current supply circuit, and when synchronism is established the main alternating-current switches are closed and the main direct-current switches opened. On account of the low resistance of the converter armature it is often a difficult matter to determine the exact instant when the proper phase relations are established, but if this has been correctly judged and the switches quickly thrown, the converter will immediately fall in step with the alternating-current circuit and run as a synchronous motor.

381. *When starting with an auxiliary induction motor, or in the manner last described what means are employed to determine when synchronism between the converter and the alternating-current supply is established?*

Incandescent lamps connected as in Fig 82 will show when the proper phase relations are established. With the lamps connected across one of the main alternating-current switches and either of the other two switches closed, the full lamp current will flow from the alternating-current circuit through the lamps and one branch of the armature winding, so long as the converter is motionless. When it is revolving, its electromotive force opposes the circuit electromotive force whenever the two are exactly in phase and agrees with it when they are opposed in phase. Consequently the lamps are lighted and extinguished alternately, as the converter speeds up, the flickering becoming slower as the converter approaches synchronous speed. At synchronism, the lamps remain dark; then the other two switches should be thrown without delay.

382. *Would a slight variation in the frequency of the supply circuit cause any noticeable effect in the operation of a rotary converter?*

It would tend to cause a varying speed of the armature with respect to the impressed voltage. This objectionable feature is called hunting.

383. *Can the hunting of rotary converters be prevented in any way?*

The tendency to hunt can be greatly reduced by increasing the inductance of the armature; this is done by placing the armature windings in deep slots having teeth with projecting heads. Hunting can also be checked by providing the pole-faces with heavy copper grids that extend across them and are imbedded in slots. The grids serve as dampers to the hunting effect, their action being analogous to that of a dash-pot on an engine governor.

384. *Can rotary converters be operated in parallel?*

Yes. If they are to be operated in parallel on the direct-current side they are connected in parallel like shunt or compound-wound direct-current generators. The load can be divided among them by regulating the alternating voltage, by varying the field current, or, to a slight extent, by changing the position of the brushes on the commutator. When converters are to be operated in parallel on the direct-current side and take their supply current from a single alternating-current system, separate transformers must be provided for each converter. A single set of transformers may, however, be used if there be separate secondaries for the converters.

385. *What should be done in case a converter flashes over or the direct-current circuit-breaker operates on account of excessive current?*

The deflection of the direct-current voltmeter needle should be noted to see that the field has not been reversed in polarity. If the polarity has not changed, the deflection will be in the usual direction. If the polarity of the field has changed, there will be no deflection, or there will be a slight one backward, and the polarity must be reversed by remagnetizing the field-magnet from the direct current circuit. After a converter flashes over, it should, if possible, be shut down for a moment and the commutator cleaned of any burs that may have resulted.

386. *In case the alternating-current circuit-breakers operate, what should be done?*

The direct-current circuit-breaker and switches should be opened, and the converter synchronized as in first starting

387. *If the power in the alternating-current circuit ceases, what should be done?*

Shut down the converter immediately, opening all the switches. The synchronizing device should then be connected up so the lamps will light and thus show when the power is again ready for starting the converter.

388. *How should a rotary converter be shut down?*

First, the direct-current circuit-breakers should be opened, thus removing the load from the machine. Then, in turn, the

direct-current switches, the alternating-current circuit-breakers and the alternating-current switches should be opened. All the resistance should be introduced in the rheostat and the synchronizing device cut out of circuit. Before the converter stops, the commutator should be wiped off. Then the brushes should be examined and the whole machine cleaned thoroughly.

389. *Can a rotary converter be used for changing direct current into alternating currents?*

Yes. It is then called an "inverted" converter. The machine is separately excited and one rheostat is required in the converter field circuit and another in the field circuit of the exciter. The rheostat in the converter field circuit is used only in starting and should be cut out when the converter is in normal operation.

390. *Does the operation of an inverted rotary converter differ from that of a rotary converter as already described?*

Yes; an inverted rotary converter does not operate at constant speed as in the previous case. Its speed depends upon the strength of its field-magnet and consequently upon the reaction of the armature current upon the field so that it varies according to the alternating-current load. If this load be a heavy inductive one, the speed may become very high. This may, however, be prevented by exciting the converter from a generator driven by the converter, so that its speed varies with that of the converter. A change in the speed of the converter will then cause a similar change in the excitation voltage and this, in turn, will produce a corresponding variation in the converter field-magnet strength. Care must be taken always to start an inverted rotary converter with load, as otherwise it is liable to run away.

391. *If a rotary converter be driven by outside mechanical power, could it be used as a generator?*

Yes. It is then called a double-current generator, because it can supply both alternating and direct currents. The capacity of a machine thus operated is no greater than that of a direct-current generator of the same total output. This is true because the heating of the armature depends on the sum of the currents in it, and not on their difference, as when the machine is operated as a rotary converter. Separate field excitation alone is advisable for a double-current generator.

392. *Are double-current generators adapted for any particular class of work?*

Yes; they are adapted for service where direct current can be employed in a certain district for a part of the day and alternating current used in some other district for another part of the day. The machine is then kept under practically constant load and takes the place of two different types of generators.

A Comparison of the Cost of Power as Generated by Different Types of Steam Engines

BY W. M. WILSON, M. M. E.

With the increase in competition in the commercial world, greater attention is given to the different items of expense. Among these items of expense, and one which ranks high in importance, is that of the cost of power. Indeed, so important is this factor that in many cases the success or failure of a commercial enterprise depends upon the price at which power can be obtained. To meet this ever increasing demand for cheap power, much attention has been given to the design of engines to see if it were not possible to decrease the steam consumption for a given power output. The result is that at the present time there are on the market a number of different types of engines, some of which have claimed for them certain advantages, and others, other advantages. This makes the choice of the type of an engine for a given installation difficult. The fact that all these engines exist in a market where competition is free goes to prove that under one given set of conditions one type is desirable, while under certain other conditions another type is best. It has been the endeavor of the writer to determine the relative merits of the different types of engines under different assumed conditions.

The work was first undertaken while the writer was a graduate student in mechanical engineering at Sibley College, Cornell University, in 1903 and 1904, and the result of the investigation was presented to the faculty of Sibley College for the advanced degree in mechanical engineering. After the work was completed it seemed that some of the general assumptions could be made to approximate more nearly the average conditions in commercial work. The work has been revised with these changes in mind and the results are given below. The computations leading to the results have for the main part been omitted from this paper; enough has been given, however, to indicate the method by which the computations have been made.

The actual running conditions of two power plants very seldom agree. One may be located in such a place that either coal or labor is more expensive than it would be in another plant of exactly the same capacity and the same type of machinery. Evidently, then, if we are to make a comparison of the different types of engines, it will be necessary to assume that all conditions which are independent of the type of the engine are constant. It is not possible to assume conditions which will agree with the actual conditions found in practice, for these vary. The writer has contented himself by assuming conditions, which, according to the best of his knowledge, represent the average conditions obtained in practice, and then stating clearly in each case exactly what these assumptions are, so that the reader can acquaint himself with the conditions upon which the computations are based and can change the results to agree with any particular case which he may have in mind.

The information leading to the different assumptions has been obtained from teachers in engineering schools and consulting engineers, from current technical literature and theses on file in Sibley College, and from manufacturers, either by correspondence or through their catalogs.

The different types of engines considered are simple non-condensing high speed, compound non-condensing high speed, compound condensing high speed, De Laval turbine, horizontal and vertical compound, condensing, low speed, and Parsons steam turbine.

A comparison of these different types was made for plants of the following capacities: 100, 200, 400, 600, 1200 and 2000 indicated horse-power.

It was assumed that in each case the engine is direct connected to a continuous-current generator. As the indicated horse-power of a steam turbine cannot be determined, the power of the turbine is expressed in terms of equivalent indicated horse-power on the basis of 1 kilowatt = 1½ indicated horse-power.

As it is essential that a plant be able to furnish power at all times, it has been assumed that in all cases one relay unit has been provided to be used in case of accident to any of the others.

Two general classes of plants have been considered: one in which the initial cost has been kept as low as possible, and another in which the initial cost has been increased in order that the running expenses might be less. In the first case it has been considered that the plant is equipped with either high-speed engines or De Laval turbines, and with fire-tube boilers and heaters. In the second case it has been assumed that the equipment consists of compound condensing, low-speed engines or Parsons turbines, water-tube boilers, heaters and economizers.

In choosing engines for any permanent installation the points most considered are reliability and expense. All the reciprocating engines enumerated above have already proven their reliability and the turbines put upon the market the last few years have proven quite satisfactory. This being the case, the choice of the type of engine to be used depends mostly upon the matter of expense, with a slight premium on the reciprocating engines because of their more thoroughly proven reliability.

The total expense of a plant is made up of two parts. One part consists of interest, depreciation, taxes, insurance and repairs, usually expressed as a per cent. of the initial cost of the plant; and the

other consists of the cost of fuel, oil, waste, and attendance.

To make a fair estimate between the different types of engines each plant should be equipped with the size of engines for which the expense will be the least. By plotting a curve for which the abscissas were the sizes of the engines in indicated horse-power, and the ordinates were steam consumption per indicated horse-power hour, it was found that for a given type of engines the steam consumption per indicated horse-power hour was the same for all sizes of engines considered in this paper, with an exception in the case of the De Laval turbine. This being the case, the choice of the size of engines for a plant in which the load is quite constant is determined by the initial cost of the plant.

The different items of the initial cost for a plant of a given capacity, which vary with the size of the engine, are the cost of the engine, generator, and building.

The cost of engines and generators was furnished in some cases by the manufacturers and in some cases by consulting engineers. The cost of the building for the horizontal engines and turbines was taken at $1.50 per square foot; and for the vertical engines at $0.11 per cubic foot. This is approximately the cost of a steel framed building having brick walls and a fire-proof roof, and does not include any foundations or smoke stacks. The price of land was taken at $0.50 per square foot.

In determining the cost of an engine room, each engine was considered as occupying a space equal to the area of its foundation, plus the surrounding space necessary to permit of its being installed and looked after while running. In the case of simple high-speed engines a clearance space of five feet wide was allowed on all sides. For tandem compound high-speed engines, five feet clearance was allowed for engines of 200 horse-power or less; above 200 horse-power five feet was allowed on two sides, and one end, and on the other end clearance as follows: 201 to 300 horse-power, five and a half feet; and from 301 to 400 horse-power six feet. For cross-compound horizontal low-speed engines, five feet of clearance was allowed on two sides and one end, and on the other end for sizes up to 400 horse-power, six feet. From 401 to 700 horse-power, seven feet, and from 701 to 1000 horse-power, eight feet. For vertical engines five feet of floor space was allowed all around, and eight feet overhead. For turbines five feet of clearance was allowed on all sides. From the prices obtained the writer determined the size of engines to be used in plants of different sizes. These are given in Table I.

The data for determining the cost of boilers, engines, and generators was obtained in part from consulting engineers, and in part through personal correspondence with manufacturers.

TABLE I.

SIZE AND NUMBER OF UNITS FOR PLANTS OF DIFFERENT CAPACITIES.

Max. Output of Plant i.h.p.	No. of Engines.	Size of Engines i h.p.
High-Speed Engines.		
100	2	100
200	3	100
400	3	200
600	4	200
1200	4	400
De Laval Turbines.		
100	2	100
200	3	100
400	3	200
600	4	200
1200	5	300
Low-Speed Engines.		
400	2	400
600	3	300
1200	3	600
2000 Vertical	5	500
2000 Horiz'tal	3	1000
Parsons Turbines.		
600	2	600
1200	2	1200
2000	3	1000

The cost of condensers, feed-pumps, and heaters is based upon estimates furnished by the manufacturers.

For all of these articles prices were obtained for as many different sizes as possible and then curves were plotted with size of units as ordinates and cost as abscissas. The price of any machine was obtained from these curves.

The cost of stacks was considered as constant for all sizes of plants and equal to $4 per boiler horse-power, for economizers $5 per boiler horse-power, and for steam piping $5 per indicated horse-power. All of the above prices are for the machines with their foundations installed ready for operation.

The prices given include one engine and one generator more than required to give the rated capacity and for duplicate feed pumps. All other parts are furnished only in sufficient capacities to give the rated capacity of the plant.

The pressures at which the engines have been considered as running are as follows: Simple engine, 100 pounds; compound condensing and non-condensing high-speed engines and De Laval turbines, 120 pounds; low-speed engines and Parsons turbines, 150 pounds. In the case of non-condensing engines it is assumed that feed water is taken from the heater at a temperature of 190 degrees, in the case of the condensing engines that it is taken from the hot well at 110 degrees, heated with heater to 160 degrees, and then in cases where economizers are used, that the latter heat it to 280 degrees.

In determining the steam consumption of the different types of engines the reports of a large number of engine tests were consulted. The different tests were made with the engine using steam at different pressures and in order to determine the corresponding consumption at the pressures assumed for the different types of engines, use was made of Dr. Thurston's formula* for the variations in the efficiency of engines with changes in the initial pressure. This formula is:

$$H = \frac{a}{\log. p},$$

where H is the number of British thermal units per indicated horse-power hour, which is to be determined.

p is the pressure of the entering steam, and a is a constant for the engine in question.

Something over a hundred tests were considered for all the types. The average steam consumptions were found to be as follows:

Simple non-condensing high-speed engines, 30 pounds per indicated horse-power hour.

Compound non-condensing high-speed engines, 24 pounds per indicated horse-power hour.

Compound condensing high-speed engines, 20 pounds per indicated horse-power hour.

Compound condensing low-speed engines, 13 pounds per indicated horse-power hour.

Parsons turbines, 13 pounds per indicated horse-power hour.

The steam consumption of the De Laval turbines is that guaranteed for different sized units. The consumption for the given sizes is shown in Table III.

In order that the paper might cover a wide field, calculations were made based upon four different prices of coal. These are $7, $5, $3 and $1.50 per ton of 2000 pounds, delivered in front of the boiler, and it is assumed that this coal has a heating value of 14,000 B.t.u. per pound. The calculation may also be made to apply to cases where the coal used has some other heating value. Take for example a coal having a heating value of 10,000 B.t.u. per pound. This coal at

$$\frac{10{,}000 \times \$7}{14\,000} = \$5$$

is the commercial equivalent of 14,000 B.t.u. coal at $7 per ton. In a similar manner the price at which 12,000 B.t.u., 10,000 B.t.u. and 8000 B.t.u. coal is equivalent to 14,000 B.t.u. coal at $7, $5, $3 and $1.50 per ton can be determined. These prices are given in Table II.

If the quality and price of the coal to be used in any case are known, the corresponding price of 14,000 B.t.u. coal can be determined from Table II.

The calculations have all been made upon a basis of a 14,000 B.t.u. coal. It is assumed that 60 per cent. of the theoretical heat in the coal is absorbed by the water in the boiler.

The cost of condensing water for the engines is taken at one cent per 10,000 pounds, and it is assumed that 30 pounds of water are required to condense one

*Thurston's "Manual of the Steam Engine," Part 1, page 878.

pound of steam. Four per cent. of the power of the engine is allowed for running the condenser.

The cost of oil and waste and of attendance have been considered as being the same for all types of engines. The cost of oil and waste has been taken at 0.033 cent per indicated horse-power for all sized plants.

In determining the cost of attendance of the different plants, estimates were made of the necessary cost of labor connected with a few different sized plants, and these were compared with the cost of attendance of actual plants in operation. A curve was then drawn with size of plants as abscissas and cost per year as ordinates. The cost of attendance of plants

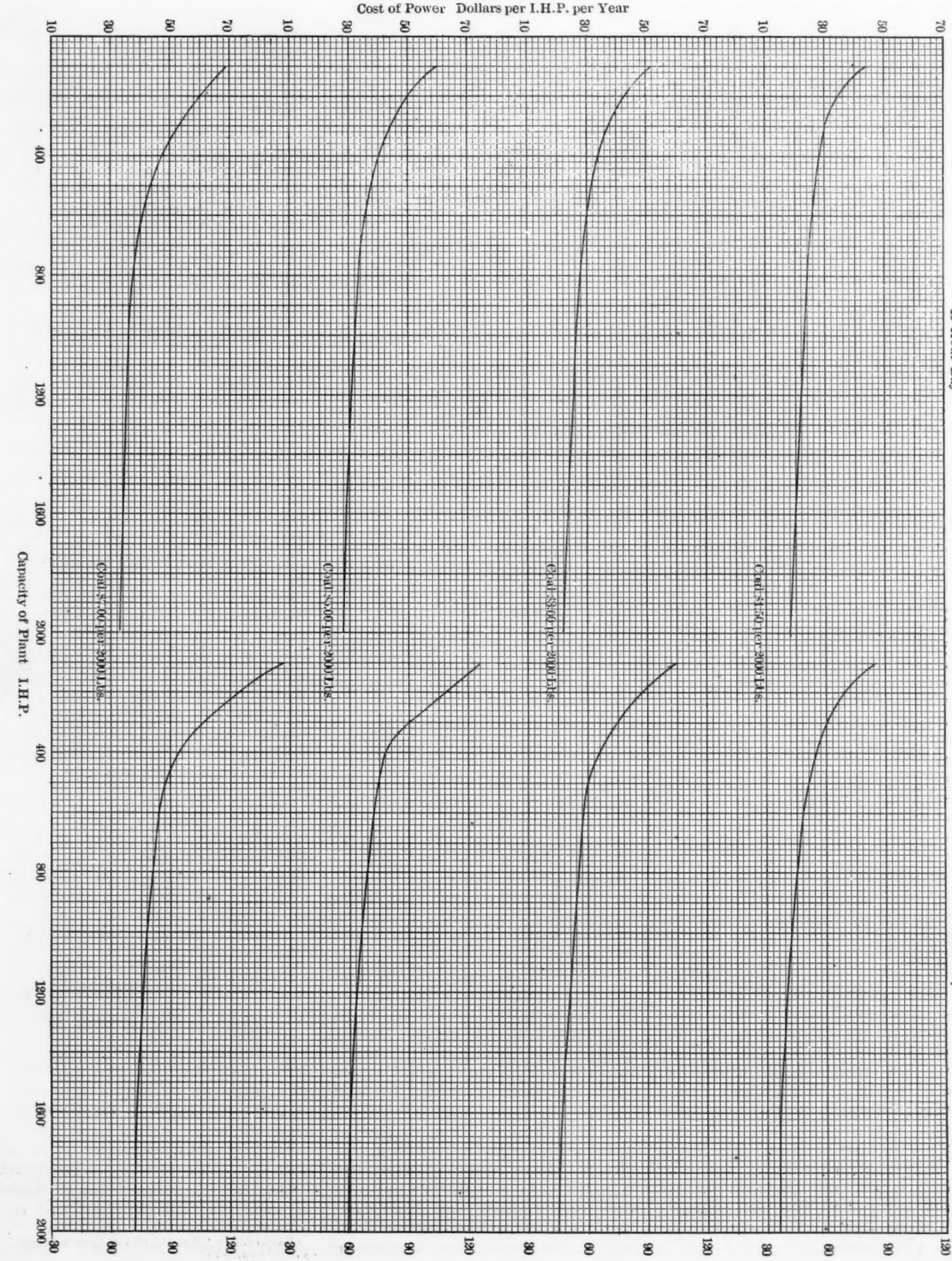

Cost of Power Dollars per I.H.P. per Year
Capacity of Plant I.H.P.
Coal $7.00 per 2000 Lbs.
Coal $7.00 per 2000 Lbs.
Coal $3.00 per 2000 Lbs.
Coal $1.50 per 2000 Lbs.
10 Hour Day
24 Hour Day

TABLE 3

COST OF POWER AS DEVELOPED BY DIFFERENT TYPES OF ENGINES

CAPACITY OF PLANT I.H.P.	TYPE OF ENGINE	STEAM PRESSURE-GAUGE	LBS. OF STEAM PER I H.P.H.R.	SIZE OF ENGINES	NO. OF ENGINES	COST OF ENGINES	COST OF GENERATORS	COST OF PIPE STEAM	COST OF CONDENSERS	COST OF ENGINE ROOM	BOILER H.P. REQUIRED.	TYPE OF BOILERS	NO. OF BOILERS	SIZE OF BOILERS	COST OF BOILERS	COST OF BOILER ROOM	COST OF FEED PUMPS	COST OF ECONOMIZERS	COST OF HEATERS	COST OF STACK	TOTAL INITIAL COST	INTEREST DEPRECIATION REPAIRS—TAXES INSURANCE	CONTINUOUS EXPENSE PER YEAR							
																							OIL AND WASTE	ATTENDANCE	CONDENSING WATER	COAL UPPER LINE 10 HR. DAY LOWER LINE 24 HR. DAY				TOTAL
																										COST OF COAL PER 2000 LBS.				COST OF COAL PER 2000 LBS.			
																										$7.00	$5.00	$3.00	$1.50	$7.00	$5.00	$3.00	$1.50
100	SIMPLE NON-COND. H. S.	100	30	100	2	2880	2300	1000		1292	92	F.T.	2	92	1904	2680	280		160	460	12956	1814	102	1550		4474	3197	1918	959	7940	6663	5384	4425
																							289	4380		11234	8024	4814	2407	17717	14507	11297	8890
	COMPOUND NON-COND. H. S.	120	24	100	2	4600	2300	1000		1430	74	F.T.	2	74	1640	2510	230		130	296	14136	1979	102	1550		3594	2568	1540	770	7225	6198	**5171**	**4401**
																							289	4380		9022	6444	3867	1933	15670	13092	10514	**8581**
	COMPOUND COND. H. S.	120	20	100	2	4600	2300	1000	1400	1430	66	F.T.	2	66	1500	2414	200		125	264	15233	2133	102	1550	193	3206	2291	1374	687	7184	6269	5352	4665
																							289	4380	546	8049	5749	3449	1725	15397	13097	10797	9073
	DE LAVAL TURBINE	120	18.6	100	2	7400		1000	1200	910	62	F.T.	2	62	1425	2360	200		120	248	14863	2081	102	1550	180	2986	2133	1280	640	**6899**	**6046**	5193	4553
																							289	4380	508	7478	5348	3209	1604	**14736**	**12606**	**10467**	8862
200	SIMPLE NON-COND. H. S.	100	30	100	3	4290	3450	1500		1830	185	F.T.	3	93	2650	3850	530		235	740	19275	2699	205	2170		8949	6392	3836	1918	14023	11466	8910	6992
																							578	6132		22468	16048	9628	4814	31877	25457	19037	14223
	COMPOUND NON-COND. H. S.	120	24	100	3	6300	3450	1500		2040	148	F.T.	3	74	2460	3600	440		200	592	21182	2965	205	2170		7188	5135	3081	1540	12528	10475	8421	**6880**
																							578	6132		18044	12888	7733	3867	27719	22563	17408	**13542**
	COMPOUND COND. H. S.	120	20	100	3	6900	8450	1500	2050	2040	132	F.T.	3	66	2250	3440	400		180	528	22738	3183	205	2170	387	6413	4581	2748	1374	12358	10526	8693	7319
																							578	6132	1093	16097	11498	6899	3449	27083	22484	17885	14435
	DE LAVAL TURBINE	120	18.6	100	3	11100		1500	1940	1270	124	F.T.	2	124	2480	2825	390		180	496	22181	3105	205	2170	360	5972	4266	2559	1280	**11812**	**10106**	**8399**	7120
																							578	6132	1016	14956	10696	6418	3209	**25787**	**21527**	**17249**	14040
400	SIMPLE NON-COND. H. S.	100	30	200	3	7020	6630	3000		2450	370	F.T.	4	123	4920	5312	1000		400	1480	32212	4510	409	3026		17898	12784	7671	3836	25843	20729	15616	11781
																							1156	8550		44936	32096	19256	9628	59152	46312	33472	23844
	COMPOUND NON-COND. H. S.	120	24	200	3	10620	6630	3000		2698	297	F.T.	4	99	3960	5098	840		336	1188	34369	4812	409	3026		14377	10269	6161	3081	22624	18516	14408	**11328**
																							1156	8550		36088	25777	15466	7733	50606	40795	29984	**22251**
	COMPOUND COND. H. S.	120	20	200	3	10620	6630	3000	3700	2700	264	F.T.	4	88	3670	4970	740		312	1056	37398	5236	409	3026	773	12825	9161	5497	2748	22269	18605	14941	12192
																							1156	8550	2187	32195	22996	13798	6899	49324	40125	30927	24028
	DE LAVAL TURBINE	120	17	200	3	18960		3000	3250	1620	224	F.T.	3	113	3390	4000	650		275	904	36049	5047	409	3026	658	10913	7795	4677	2339	20053	16935	**13817**	11479
																							1156	8550	1858	27373	19552	11731	5866	43984	36163	28342	22477
	VERTICAL COND. L. S.	150	13	400	2	20400	10600	4000	2600	4300	153	W.T.	2	153	6100	2300	500	665	200	612	52277	7318	409	3026	503	7420	5300	3180	1590	18676	16556	14436	12847
																							1156	8550	1421	18615	13296	7978	3988	37060	31741	26423	22433
	HOR. COND. L. S.	150	13	400	2	13600	15400	4000	2600	5440	153	W.T.	2	153	6100	2300	500	665	200	612	51417	7198	409	3026	503	7420	5300	3180	1590	**18556**	**16436**	14316	12727
																							1156	8550	1421	18615	13296	7978	3988	**36940**	**31621**	**26303**	22313
600	SIMPLE NON-COND. H. S.	100	30	200	4	9360	8840	4000		3190	555	F.T.	6	111	6660	7670	1450		560	2220	43950	6153	614	3813		26848	19177	11506	5753	37418	29747	22076	16323
																							1734	10775		67404	48144	28884	14442	86056	66796	47536	33094
	COMPOUND NON-COND. H. S.	120	24	200	4	14100	8840	4000		3512	445	F.T.	5	111	5550	6440	1200		400	1780	45942	6432	614	3813		21565	15404	9242	4621	32424	26263	20100	**15480**
																							1734	10775		54132	38665	23199	11600	73073	57606	42140	30541
	COMPOUND COND. H. S.	120	20	200	4	14160	8840	4000	5050	3512	397	F.T.	5	99	4950	6296	1070		420	1588	49886	6984	614	3813	1161	19238	13733	8245	4122	31810	26305	20817	16694
																							1734	10775	3279	48292	34494	20697	10348	71064	57266	43469	33120
	DE LAVAL TURBINE	120	17	200	4	25280		4000	4500	2094	339	F.T.	4	113	4520	5240	930		380	1356	48300	6762	614	3813	986	16370	11693	7016	3508	28545	23866	19191	15683
																							1734	10775	2789	41059	29328	17597	8799	63118	51387	39656	30850
	VERTICAL COND. L. S.	150	13	300	3	29550	12600	4500	3600	5220	230	W.T.	2	230	8460	2530	650	1150	280	920	69465	9725	614	3813	755	11130	7950	4770	2386	26037	22857	19677	17293
																							1734	10775	2132	27923	19945	11967	5983	52289	44311	36333	30349
	HOR. COND. L. S.	150	13	300	3	19800	19920	4500	3600	7060	230	W.T.	2	230	8460	2530	650	1150	280	920	69490	9729	614	3813	755	11130	7950	4770	2386	26041	22861	19681	17297
																							1734	10775	2132	27923	19945	11967	5983	52293	44315	36337	30353
	PARSONS TURBINE	150	13	600	2	35000		6000	3600	1470	230	W.T.	2	230	8460	2530	650	1150	280	920	60060	8408	614	3813	755	11130	7950	4770	2386	**24720**	**21540**	**18360**	15976
																							1734	10775	2132	27923	19945	11967	5983	**50972**	**42994**	**35016**	**29032**
1200	PIMPLE NON-COND. H. S.	100	30	400	4	15200	18000	8000		4030	1119	F.T.	10	125	12300	12776	2500		1060	4440	78306	10963	1227	5580		53694	38353	23012	11506	71464	56123	40782	29276
																							3469	15768		134803	96288	57768	28884	165008	126488	87968	59084
	COMPOUND NON-COND. H. S.	120	24	400	4	22280	18000	8000		4540	890	F.T.	8	127	10160	10340	2120		860	3560	79860	11180	1227	5580		43129	30807	18455	9242	61116	48794	36472	27229
																							3469	15768		108263	77331	46398	23200	138680	107748	76815	53617
	COMPOUND COND. H. S.	120	20	400	4	22280	18000	8000	6100	4540	794	F.T.	8	113	9040	10128	1950		780	3176	85994	12039	1227	5580	2321	38476	27465	16468	8245	59643	48632	37656	29412
																							3469	15768	6558	96564	68989	41393	20697	134418	106823	79227	58531
	DE LAVAL TURBINE	120	16.6	300	5	45000		7500	7200	2960	660	F.T.	7	110	7700	8880	1700		660	2640	84240	11794	1227	5580	1925	31965	22830	13699	6850	52491	43356	34225	27376
																							3469	15768	5440	80207	57290	34374	17187	116678	93761	70845	53658
	VERTICAL COND. L. S.	150	13	600	3	43200	21300	9000	6100	7132	460	W.T.	3	230	12690	3660	1220	2300	480	1840	108992	15249	1227	5580	1509	22260	15900	9540	4772	45825	69465	33105	28337
																							3469	15768	4264	55846	39890	23934	11969	94596	78640	62684	50719
	HOR. COND. L. S.	150	13	600	3	28800	28020	9000	6100	8420	460	W.T.	3	230	12690	3660	1220	2300	480	1840	102530	14354	1227	5580	1509	22260	15900	9540	4772	44930	38570	32210	27442
																							3469	15768	4264	55846	39890	23934	11969	93701	77745	61789	49824
	PARSONS TURBINE	150	13	1200	2	45000		12000	6100	2800	460	W.T.	3	230	12690	3660	1220	2300	480	1840	88090	12333	1227	5580	1509	22260	15900	9540	4772	**42909**	**36549**	**30189**	**25421**
																							3469	15768	4264	55846	39890	23934	11969	**91680**	**75724**	**59768**	**47803**
2000	VERTICAL COND. L. S.	150	13	500	5	60000	31350	12500	8450	10735	768	W.T.	4	256	18400	4900	1920	3840	760	3072	155927	21830	2046	7440	2515	37100	26500	15901	7952	70931	60331	49732	41783
																							5782	21025	7106	93076	66483	39890	19943	148818	122295	95632	75685
	HOR. COND. L. S.	150	13	1000	3	48000	34650	15000	8450	9320	768	W.T.	4	256	18400	4900	1920	3840	760	3072	148312	20764	2046	7440	2515	37100	26500	15901	7952	69865	59265	48666	40717
																							5782	21025	7106	93076	66483	39890	19943	147752	121159	94566	74619
	PARSONS TURBINE	150	13	1000	3	62400		15000	8450	3360	768	W.T.	4	256	18400	4900	1920	3840	760	3072	122102	17094	2046	7440	2515	37100	26500	15901	7952	**66195**	**55595**	**44996**	**37047**
																							5782	21025	7106	93076	66483	39890	19943	**144082**	**117489**	**90896**	**70949**

of intermediate sizes was then read from the curve. Continuous expense based upon the original cost of the plant was assumed to be 14 per cent. per year. It is made up of the following items: Interest, 5 per cent.; depreciation, 5 per cent.; repairs, 2½ per cent.; insurance, ½ per cent., and taxes, 1 per cent.

Depreciation at 5 per cent., with interest at 5 per cent., corresponds to a life of the plant of 14 years.*

The total continuous expense per year includes the above items and in addition, the cost of coal, oil and waste, water and attendance. These items are given in detail, together with their sum, in Table III

TABLE II.

B.t.u. Per Lb. of Coal.	Price Per Ton.			
14,000	7.00	5.00	3.00	1.50
12,000	6.00	4.30	2.57	1.28
10,000	5.00	3.57	2.14	1.07
8,000	4.00	2.86	1.71	0.86

below. It is the total continuous expense per year given in that table which is used as a basis for comparing the different types of engines.

The portion of time which the engine is in operation affects the ratio between the expenses due to original cost and running expenses. With this in mind, two cases have been considered, one in which it is assumed that the engines run at rated capacity for ten hours per day, 310 days per year, and the other in which it is assumed that they run at their rated capacity 24 hours per day, 365 days per year. When the plant is used only ten hours per day 310 days per year, coal is required for banking fires and getting up steam before the ten-hour period that the plant is in operation. An allowance of five pounds of coal per boiler horsepower per day has been allowed for this purpose. No such allowance is necessary in the 24-hour day. Attention is called to the fact that, as stated above, the engines are considered as being run at their rated capacity. This the writer feels is justified by the fact that one relay unit is provided in each case, which can be brought into action in case of an unexpected increase in load. To any who wish to make the comparison on the basis of a load factor other than 100 per cent., attention is called to Professor Carpenter's paper in the Trans. A. S. M. E., Volume XIV, page 474, in which he gives the variation in steam consumption of reciprocating engines with variations in load factors; and also to a paper by E. H. Sniffen, published in the Proceedings of the American Street Railway Association, 1902, giving the same data for Parsons turbines, and to "The De Laval Steam Turbine," published by the De Laval Steam Turbine Company, New York.

With the data relating to initial cost given in this paper, and to the variation in steam consumption with variations in the load factor as given in the papers referred to, a comparison can be made of the different types of engines for a plant of a given capacity and for any assumed load factor.

In looking over Table III it is seen that a number of factors are involved in determining the choice of the type of engine for a given installation. These are given in tabular form below.

Which of the above items are of the most importance depends upon the conditions pertaining to each particular plant and could only be determined by making computations similar to those made in obtaining the results given in Table III.

In Table III the lowest figures for the total continuous expense for each sized plant are in black type.

Referring to Table III it is noticed that of the different types of engines, the simple high-speed and compound condensing high-speed engines appear at a disadvantage under the conditions assumed. This is due in the first case to the large steam consumption and in the latter case to the cost of condensing water. In a plant where the load factor is smaller and the portion of time that the engine is in operation is smaller, the simple engine makes a much better showing. The noticeable feature of Table III is the advantage at which the turbines appear, the De Laval for the smaller plants, and the Parsons for the larger plants.

In analyzing the cost it is seen that this advantage of the turbines is due to the low cost of the turbines themselves as compared with the cost of the engines, and also because the cost of the generators run with the turbines is much less than it is for those which run with the engines, because of the higher speed at which they are run.

On the accompanying chart are curves showing the variation in the cost of power with different sized plants. The cost as given makes no allowance for the cost of the switch-board, electrical connections, or lighting and heating the building. These items are independent of the type of engine, so do not affect the main object of the paper; however, they do affect the curves shown on Plate I. Each curve should be raised somewhat to give the total cost of the power. However, the change would be slight and would change but very little the general direction of the curves, so that the curves do give approximately the variations in the cost of power with variations in the size of the plant.

Type of Engine.	Advantages.	Disadvantages.
High-Speed.	Low initial cost of engine. Moderate cost of generator. *Cheap type of boilers. Moderate floor space.	Large coal consumption. Large boiler capacity.
Low-Speed.	Low coal consumption. Small boiler capacity required.	High initial cost of engines. High initial cost of engines. *Expensive type of boiler. Large floor space.
Parsons Turbines.	Moderate initial cost of turbine & generator. Small floor space. Small boiler capacity. Low coal consumption.	Expensive type of boiler.
De Laval Turbines.	Moderate initial cost of turbine & generator. Small floor space. Moderate boiler capacity. Moderate coal consumption. Cheap type of boiler.	
Condensing Plants.	Decreased coal consumption. Decreased boiler capacity required.	Initial cost of condenser. Cost of condensing water.

*Where the boiler pressure does not exceed 120 pounds, fire-tube boilers can be used, but with the higher pressure it is customary to use the more expensive water-tube boilers.

Heating of Electrical Apparatus

In specifying temperature rises of 70 to 80 degrees Fahrenheit for electric generators and motors, engineers are no doubt governed by the consideration that the cotton covering on wire begins to carbonize at about 255 degrees Fahrenheit, and that internal maximum temperatures are higher than those indicated by the thermometer. Even though the insulation be of mica, or other heat-resisting material, the dynamo is essentially a piece of apparatus the temperature of which must be kept low. But a rheostat, though not always used directly with that intention, is an appliance for dissipating energy. When properly designed, it has nothing whatever inflammable in its construction and it should be so placed that there is nothing inflammable in proximity.

Cassier's Magazine has been sold by the estate of the late Louis Cassier to Henry Harrison Suplee, for many years the technical editor of the *Engineering Magazine*, and James Van Vorst Colwell, recently works manager for the C. W. Hunt Company, and the publication will be continued as heretofore by the Cassier Magazine Company, at No. 3 West Twenty-ninth street, New York. The editorial conduct of the magazine will be in the hands of Mr. Suplee, who brings to this work a wide experience both in the practice of mechanical engineering and of technical journalism, while the business department will be directed by Mr. Colwell. The first issue of the magazine under the new management will be the January number.

*See Turneaure and Russell, "Public Water Supplies," page 209.

A Study of the Zeuner Slide-Valve Diagram

BY E. S. HAWKINS

Probably every engineer at some time or other has had a nice job of valve setting on his hands. In fact it is quite common to hear an engineer begin a conversation with some such remark as this: "Well, the worst job I ever had was to set the valves on an old high-speed engine with a riding cut-off valve." In another case it may be an experience in setting the valves of a double-eccentric Corliss or, perhaps, in still a third case it was a tandem or cross compound engine. Wherever one goes, he is sure to meet an engineer who is on the lookout for a patent-medicine cure-all that will be a sure preventer of the troubles met in setting valves. Perhaps this may not appear unusual when one considers the fact that on the proper or improper setting of the valves may depend the balance of profit or loss when it comes to the final reckoning at the end of the year.

For those who have mastered the principles of valve operation, the work of setting the valves is simple, but to a large number of engineers the functions of the valve seem to be hidden behind the valve-chest cover. Even with the cover removed and the valve parts in plain sight, it is not always clear as to just what changes are necessary to obtain a desired result. In some cases, it is possible to tell at a glance where the trouble lies, but in others one must make a careful study of the case, considering the movements of the valve in their relation to the engine piston.

The making of this analysis is not an easy matter for those unfamiliar with mathematical problems and, although many men can by a few figures or rule-of-thumb methods determine what the valve conditions should be, there are others for whom the task is not so simple. To these men, objects, lines and dimensions that are visible to the eye make the subject much clearer than could be done by the mazes of mathematics only suitable for exploration by the logically trained analyst. It is, therefore, with a view to meeting the needs of the practical man, to whom the chalk sketch tells everything, that this article is written.

Several methods of graphically solving slide-valve problems have been worked out, but Zeuner's diagram is, perhaps, the most useful and simple of all and will, therefore, be considered in detail. By means of the diagram, the effect on the steam supply of changes in the dimensions of the valve, or in the adjustment of the eccentric, etc., are made actually visible.

It is assumed that the length of the eccentric rod is infinite or, in other words, that it moves in a line parallel to the piston rod. This assumption can safely be made, since the angle due to the eccentric radius is so small as to have no appreciable effect on the results. Given the points at which admission, cut-off, etc., take place, the diagram shows the relative crank position. Conversely, the position of the valve corresponding to a given crank position may be found readily. By the use of the valve diagram in connection with the indicator, it is possible to diagnose a case of valve trouble so as to know the dimensions of the valve and the way in which it performs its functions as accurately and certainly as if it were seen operating within a transparent chest.

For the sake of clearness, let it be supposed first, that the valve has no lap and that the line *O*—*Q*, as in Fig. 1, represents the eccentric radius, so that the eccentric travels in the large circle as indicated by the arrow. When it is at *Q*, the valve is closed but as it moves around the circle the valve gradually opens until, when the eccentric is at the point H, the amount of opening is represented, to the scale of the diagram, by the distance *O*—*G*. It will be noted here that the crank positions are not shown, but for any given position of the eccentric in its circle, the corresponding position of the crank will be behind the eccentric position by 90 degrees plus the angle of advance.

Since it is desirable to avoid the necessity of dropping perpendiculars to the line of valve travel, such as *H*—*G* in Fig. 1, it becomes necessary to construct a line each point of which is distant from the center *O* by an amount equal to the valve travel corresponding to the eccentric position indicated by a line passing from *O* through the given point. To do this, take the point *O* as a center and, with *O*—*G* as a radius, draw the arc *G*—*L* until it intersects the line *O*—*H*. Then at the position *H'* of the eccentric, drop the perpendicular *H'*—*G'* to the line *X*—*O*—*Y* and, with *O*—*G'* as a radius, draw another arc intersecting the line *O*—*H'*. By continuing this process, a number may be found each of which marks off on the line of eccentric position a distance from the point *O* equal to the valve travel for the given position of the eccentric. Upon joining the points, such as *L* and *L'*, a closed curve is formed known as the valve circle.

To prove that the curve is a circle, take the triangles *O*—*L*—*Y*—*O* and *O*—*G*—*H*—*O*, as in Fig. 1, in which *O*—*Y* equals *O*—*H* and *O*—*G* equals *O*—*L*, while the angle *L*—*O*—*Y* is common to both triangles. Since the two triangles have two sides and an included angle equal the one to the other, they can be placed in coincidence and are, therefore, equal the one to the other, so that the line *H*—*G* equals the line *L*—*Y* and the angle *O*—*L*—*Y*, being the same as angle *O*—*G*—*H*, is a right angle. The right angle *O*—*L*—*Y* being inscribed in the closed curve and on its diameter *O*—*Y*, gives proof that the curve is a circle.

Perhaps it may be thought that the proof of the valve circle is long and unnecessary, but it is important to establish this fact since, after having done so, the circle can be drawn in on the eccentric radius as a diameter without plotting the points as at *L*. Thus the diagram is simplified and the work shortened. By reasoning similar to what has been given, the valve circle for the right-hand end of the valve can be drawn in at the left of the figure, as in Fig. 1; and, since the valve is supposed to have no laps, the two valve circles will be identical, that at the right of the figure representing the opening of the valve for the admission of steam at the left while, at the same time, the circle at the left of the figure represents the opening of the valve at the right hand for the escape of the exhaust steam. On the reverse stroke of the piston, what was before the circle for steam admission is that for exhaust and *vice versa*. Thus in Fig. 1, for the eccentric position *J*, the valve will be open for the admission of steam at the right-hand end by the distance *O*—*K*.

In practice, it is not possible to have a valve with no lap, because if such a valve were constructed it could not be closed tight enough to prevent the escape of steam past the edge of the port; nor could the periods of admission and exhaust be regulated to secure the highest economy. Having, therefore, drawn in the valve circle in Fig. 2, by the process already described, it will be necessary to take account of the lap on the two sides of the valve, by drawing in the arcs *C*—*D* and *A*—*B* with centers at the point *O*. The distance *O*—*C* represents, to the scale taken, the amount of the steam or outside lap of the valve, while the distance *O*—*A* represents the inside or exhaust lap. When, therefore, the eccentric is at the position *W*, Fig. 2, the amount the valve has opened at the left for the admission of steam is represented by the distance *G*—*H*, or the sector *O*—*H* less the length of the steam lap *O*—*G* by which amount it is necessary for the valve to travel before the port opens. Similarly, the distance *I*—*J* at the left of the diagram is the amount by which the valve is open for the exhaust of steam on the right-hand end for the same position *W* of the eccentric.

Usually it is most convenient to refer all valve motions to the position of the crank, since it is easier to determine this position than that of the eccentric and, for this reason, it is now necessary to translate the eccentric positions, discussed in Figs. 1 and 2, to the corresponding crank positions. This is done in Fig. 3 by moving the diameter of the valve circles backward by an amount equal to 90 degrees plus the angle of advance.

Suppose that the crank is on the inner dead center, as at *C* in Fig. 3, the corresponding position of the eccentric is at *P* so that the angle *P*—*O*—*R* is the angle of advance. This being the case, the distance *O*—*A* represents the amount of valve

travel from the mid position at the beginning of the stroke. As the crank revolves to the positions C_1, C_2, E_3, etc., the eccentric moves through equal angles to the corresponding positions P_1, P_2, P_3, etc., until, at this latter point, it has had its maximum effect on the motion of the valve which has now reached the full limit of travel to the right. At this time, the crank is evidently at the point E_3 so that

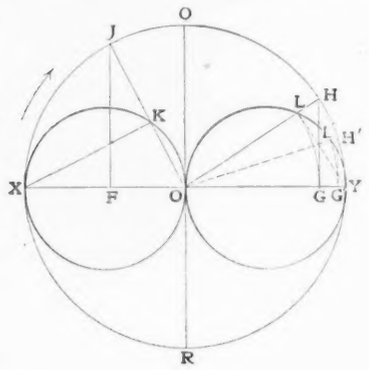


FIG. 1

the angle $R—O—E_3$ equals the angle $P—O—R$, or the angle of advance. The distances $O—A$, $O—A_1$, $O—A_2$ and $O—P_3$ represent the travel of the valve from its mid position for the various crank positions C, C_1, C_2, and E_3. By sweeping circles from the points A, A_1, A_2, etc., to the corresponding points E, E_1, E_2, etc., with O as a center, and connecting E_3 with any one of the points, such as E_1, we have a second method of proving the fact that the valve curve is a circle, for, as can readily be seen in the figure, the triangle $O—E_1—E_3$ is equal to the triangle $O—A_1—P_1$.

As may be inferred from what has already been said, the amount that the valve overlaps the edge of the port when in the mid position is called the "lap," the part on

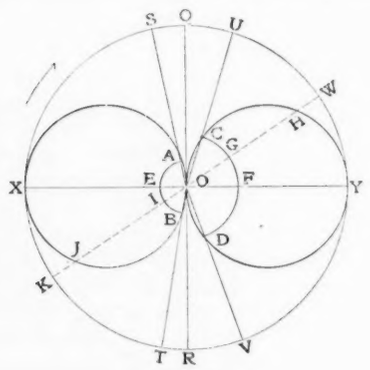


FIG. 2

the outside of the valve being known as the steam or outside lap while that on the inside is the exhaust lap. Before the steam port can open, the valve must travel a distance equal to the amount of the lap and, after it has opened, the amount of opening is always equal to the amount of travel less the amount of lap. This fact has been shown diagrammatically in Fig. 2 and is also shown in Fig. 4, in which it will be seen that for the position K the opening of the port is zero. This point, therefore, represents the crank position at the instant the valve is ready to open or, in other words, the point of admission. It is evident from the diagram that at the point P the opening of the valve is again zero, so that this point is the position of the crank when the valve closes and cuts off the steam supply.

When the crank is on the dead center, as at C in Fig. 4, the valve is open by the amount of lead, which is represented in the diagram by the distance $N—Q$ on the line of dead centers. In a similar way, the amount that the valve is open to exhaust at the end of the stroke is known as the exhaust lead.

It is now necessary to show the proof of some of the more important properties of the diagram as shown in Fig. 4. By examining the figure it will be seen that a perpendicular let fall from the point C_3, where the valve-circle diameter intersects the crank circle, cuts the line of dead centers at a point N distant from the center O by an amount equal to the lap plus the lead of the valve. To prove that the line joining the points of admission and cut-off is tangent to the steam-lap circle, take the triangles $C_3—O—L$ and $P—O—M$, of which the sides $O—P$ and $O—M$ are equal respectively to the sides $O—C_3$ and $O—L$, and the angle $C_3—O—P$ is common to both of the triangles. Hence they are equal and the angle $O—M—P$ equals the angle $O—L—C_3$, which is a right angle, thus proving that the line $P—M—K$ is tangent to the circular arc $Q—M—L$. From this proposition it follows that if a circle be drawn with the point C as the center and a radius equal to the lead, this circle will be tangent to the line $K—P$, which joins the points of admission and cut-off. As a proof, the line $C—X$ is drawn from C parallel to $K—P$, and therefore perpendicular to $C_3—O$, so that the lines $C—O$ and $C_3—O$ are equal, as are also the angles $O—C—X$ and $O—C_3—N$. Since the angle $C—O—C_3$ is common to both triangles, they are equal and $X—O = N—O$, while $X—M = N—Q$ = the lead. Lines $C—X$ and $M—R$ are parallel by construction, so that $C—R = X—M$ = the lead, and $C—R—M$ is a right angle with the circle tangent to the line $K—P$.

Another property that is useful in constructing the exhaust part of the diagram when the lap circle is so small that it does not give a sharp intersection with the valve circle, is the fact that if $O—F$ be drawn perpendicular to $O—C_3$, and a perpendicular be let fall from the point F on the admission line, it will be equal in length to the radius of the steam-lap circle. The truth of this proposition is readily seen from the fact that the triangles $F—O—Y$ and $O—K—M$ are equal, because they have equal hypotenuses and are right-angled triangles with the angles $F—O—Y$ and $M—K—O$ equal. Lines $F—Y$ and $O—M$ being similar sides of equal triangles are themselves equal.

To construct the part of the diagram corresponding to the exhaust side of the valve, the reasoning is the same as that just given for the steam side, the piston being, however, on the return stroke from P_3 to C, in Fig. 3, so that the valve circle comes in the lower half of the figure. The angle of advance and travel of the

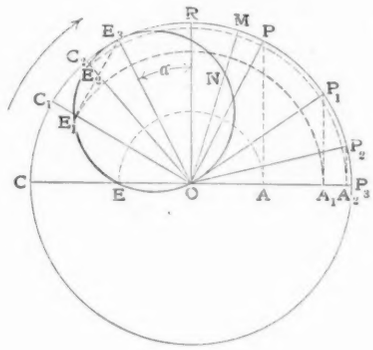


FIG. 3

valve are the same as for the steam side of the diagram, but the lap is somewhat less, since economical performance requires as small a lap as possible, and the lower steam pressures usually prevailing on this side of the valve make it possible to have a tight seating valve without a great amount of lap. The exhaust lead and the points of release and compression correspond respectively to the steam lead and the points of admission and cut-off, and are obtained on the exhaust-valve circle in the same manner as the corresponding points of the steam-valve circle.

Fig. 5 shows a complete diagram, in which admission takes place at A, as represented by the angle $A—O—X$, while cut-off takes place at C, as given by the angle $C—O—X$. In this figure, the steam

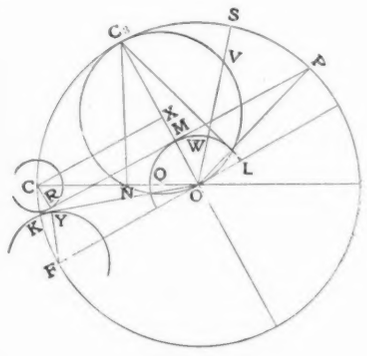


FIG. 4

lap and steam lead are represented respectively by the distances $O—N'$ and $N'—N$.

In practical construction, it is not possible to have a steam port wide enough to give full opening to the exhaust, so that the width of this port must be marked off, as in the figure, by the arc $W—W'$, the length of the line $L—W$ representing to scale the width of the port so that the

exhaust port is open wide when the crank is in the position *O—W*, and remains open until it reaches the position *O—W'*. If this same width of the steam port be laid off in addition to the lap on the diameter of the steam valve circle, the point for the port to be wide open will fall at *E'* in the figure, thus showing that this port can never be fully open since the distance *O—E'* that the valve would have to travel is greater than its maximum displacement.

From this it is to be seen that when the valve has moved to give the steam port its maximum opening, the left-hand end of the valve overlaps the right-hand end of the port by an amount equal to the distance *E—E'* in the figure. Fig. 5 is the diagram for the head end of the cylinder, the diagram for the crank end being similar, with the exception that the laps are different.

In Fig. 5 is shown the probable form of indicator diagram that would be obtained from an engine represented by the valve diagram shown. The points of admission, cut-off, dead center, release, compression, etc., of the valve diagram are projected down to the corresponding points of the indicator diagram. A base line is then established, and the probable line of vacuum is laid off on a vertical scale of pressures on

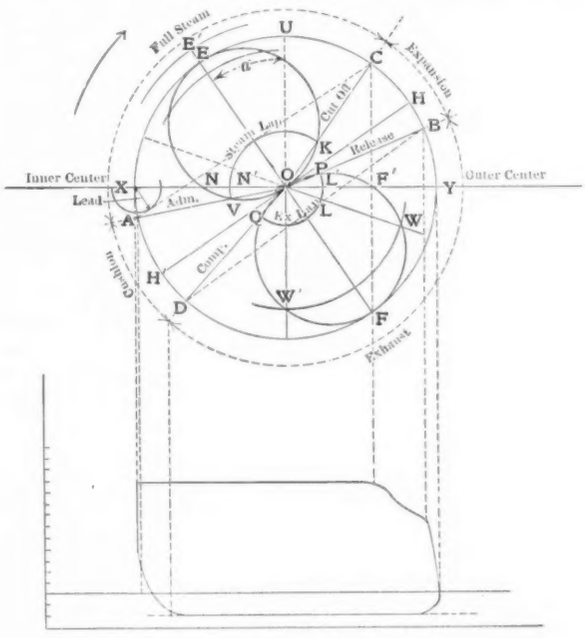


FIG. 5

which the boiler and admission pressures are also laid off at proper hight for the assumed boiler pressure available. In this way, the points where the curvature of the diagram boundary changes are determined, and smooth curves are sketched in by eye to make connections.

Diagrams of this kind are of service principally in designing the engine, but they may be of considerable use in every-day work, by showing approximately what shape the diagram, taken with the indicator, should have. This is especially the case with high-speed engines with which it is often difficult to know whether or not the diagram taken with the indicator represents the actual performance of the engine. In some cases there are forces at work which distort the diagram of the high-speed engine until it is difficult to recognize the different valve operations. Sometimes the freaky diagrams are the result of irregular expansion and contraction of the valves, ports and steam passages, with the engine at different temperatures, as after running some time or when just starting up.

Altogether, there are twelve points of information given by the kind of valve diagram shown in Fig. 5, those not already mentioned being the angle of advance *E—O—U*; the travel of the valve *X—O—Y*; release at *B*, given by the angle *B—O—Y*; compression at *D*, given by the angle *D—O—X*; the exhaust lap *O—L*; the exhaust lead *F'—L'*; and the maximum opening.

In any consideration of the valve by the diagram, it is necessary that the points of admission and cut-off, as well as the point of compression and the exhaust lap, be given in terms such that they may be laid down on the diagram by angular distances of the crank from the beginning or end of the stroke. If, as is sometimes the case, these items of information are given as taking place at a certain part of the stroke, it will be necessary to find the angular positions corresponding to the data given by drawing a crank circle or a circle of reference on a new scale. In doing this, it is most convenient to draw the circle concentric to the valve diagram.

As an example, suppose that the cut-off is given as at five-eighths of the stroke; the crank circle is then drawn as in Fig. 6, so that the distance *C—P*$_3$ represents, to the selected scale, the stroke of the engine. The distance *C—A* is then laid off equal to five-eighths of *C—P*$_3$ and with a radius corresponding, by the selected scale, to the length of the connecting-rod and with the center at *B*, an arc is swept from *A* to *P*. The angle *C—O—P* thus formed is the angular distance from the beginning of the stroke at which cut-off takes place. This angle is then transferred to the valve diagram and any remaining angular distances still unknown

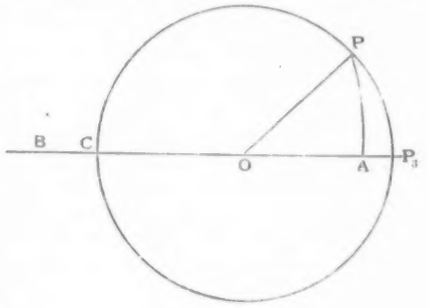


FIG. 6

are found and transferred in the same manner.

It will be noted that the different points of information given by the diagram appertain part to the steam side and the others to the exhaust side of the valve, while the angle of advance and the valve travel are common to both sides. In order to construct a complete diagram, it is necessary that four of these points of information be given, although for the construction of either the steam or exhaust side alone three points will be sufficient. Where the complete diagram is to be constructed, one or two of the points given must belong to a different side of the diagram from the others, and one of the points must be a linear measurement, such as the steam lap, steam lead, or maximum opening; otherwise, as can readily be seen, the angular measurements could be used to construct a valve of any given linear dimensions provided only that the different measurements bear a certain relation to each other. Since the opening to exhaust is greater than that to steam, and since the ports can only be made sufficiently wide to allow full opening, the width of port when given as part of the data in any problem may be taken as the maximum opening of the exhaust port.

Storing coal in pits capable of being flooded has been adopted at the new plant of the Western Electric Company, at Hawthorne, Ill. A plot about 320x75 feet has been excavated to a depth of about 12 feet, and lined and sub-divided by concrete walls into twelve 80x25-foot pits. Their bottom is the clay sub-soil and the walls are carried about four feet above ground. The pits can be flooded by means of a 12-inch water main. The longitudinal division walls are wide enough to carry the tracks on which the coal is delivered. It is removed from the pits by a steam shovel.

THE HYDRAULIC ELEVATOR—II

Counterbalancing—The High-Pressure System—The Plunger Elevator

BY WILLIAM BAXTER, Jr.

In the preceding article of this series it was stated, in discussing the water-balance elevator, that the water used is nearly proportional to the loads carried. This statement was intended to apply to the total work done; that is, to an all-day run. It is not true, however, with reference to a single trip, either up or down. On any up trip the amount of water that must be let into the bucket depends not only upon the load but also upon the amount of water remaining in the bucket from the previous trip. On the down trip the amount of water that must be let out of the bucket depends on the amount it contains as well as upon the load, and is small if the load is great, and large if the load is light, so that it is inversely proportional to the load instead of directly proportional. For an all-day run, however, the statement as made in the previous article is substantially correct.

Looking at Fig. 9 and keeping in mind the explanations given last month, it can easily be seen that the hydraulic cylinder acts only to lift the car and that the latter is lowered by its own weight combined with that of the load. This being the case, it is evident that the empty car must be heavy enough to pull up the piston, the piston rods, the traveling sheave and frame and, in addition, to overcome all the friction of the moving parts. As a matter of fact, the weight of the car is more than enough to do this work in almost every case, so that if the car could be made lighter, power could be saved, because there would be less dead load for the hydraulic cylinder to lift on the upward trips. It is not practicable in most cases to reduce the weight of the car, but the same result is accomplished by adding a counterbalance weight on the side of the lifting cylinder, as shown at W in Fig. 9.

When the counterbalance is located in the frame of the traveling sheave, as shown in this diagram, its weight must be much greater than that of the proportion of the car weight it is intended to balance. The difference between the two weights will depend upon the gear of the elevator. If the gear is two-to-one, as in Fig. 9, the counterweight will have to weigh twice as much as the proportion of the car weight that it balances. If the gear were three-to-one, the counterweight would weigh three times as much, and for a four-to-one gear, four times as much as portion of the car it balances. From this it can be seen that the weight of the counterbalance could be greatly reduced if it were connected directly with the car by means of ropes running over an overhead sheave, as shown in Fig. 10, and this construction is commonly used, although the whole of the counterbalancing is very rarely done in this way.

A counterweight connected as shown at w in Fig. 10 is called an independent counterbalance, and it possesses some advantages as well as some disadvantages. One advantage of an independent counterbalance is that it saves weight, particularly in high-gear machines, as each pound in the counterweight will balance one pound of car, while in the counterweight W, the number of pounds required to balance one pound of car is equal to the gear of the machine; in Fig. 10 it would be four pounds.

Another advantage of the independent counterbalance is that the ropes that connect the weight with the car, carry a portion of the weight of the latter equal to the weight of the counterbalance, and thus take off this amount of strain from the main lifting ropes, and thereby render the elevator that much safer.

The principal objection to the independent counterbalance is that it interferes with making quick stops of the elevator. This objection is not very great in the case of slow-running cars, but increases rapidly as the car speed increases. Looking at Fig. 10 it can be seen that if the car were stopped suddenly when running up, the momentum of the weight w would tend to keep it running down, and thus draw up the car and slacken the ropes passing over the sheave A. It can also be seen that if the weight w were nearly as heavy as the car, the tendency to continue on the downward run would be much greater than if its weight were only a small portion of that of the car. The momentum increases as the square of the velocity, so that if the car speed be doubled, the momentum of the counterweight will be quadrupled; hence, the higher the car speed, the greater the objection to putting a large portion of the weight in the independent counterbalance.

Any weight used as a counterbalance, whether located at w or at W, will oppose the stopping of the car on the upward trip, but unless the weight is equal to a large proportion of that of the car, this opposition will amount to nothing in practice, because the distance in which a car can be stopped without discomfort to the passengers is much greater than that through which it would be carried by the momentum of a light counterbalance. The object of the counterbalance is to save power by reducing the weight that the lifting cylinder has to raise, and this object can be more fully realized by placing the weight at W than by placing it at w, because in this way the weight of the empty car can be more nearly balanced. Why this is so can be made clear by a simple calculation.

Suppose that the car in Fig. 10 weighs 1500 pounds, and that if all but 500 pounds of this weight be balanced, this remainder will be sufficient to run the car down at the desired velocity. Now if all this counterweight be put in an independent weight, as at w, we will require 1000 pounds, and if it be put in the sheave frame, as at W, we will require 4000 pounds. Suppose that the car speed is four feet per second; then, since the momentum is proportional to the square of the velocity, the momentum of w would be equal to $1000 \times 4 \times 4 = 16{,}000$, while, with the counterweight at W the velocity would be reduced to one-fourth, or one foot per second, so that the momentum would be proportional to $4000 \times 1 \times 1 = 4000$. This momentum is exerted upon the bights of the rope that pass under the traveling sheaves B and C, so that the force acting to lift the car is proportional to 1000, as compared with 16,000 in the other case.

As the force tending to lift the car, due to the momentum of the counterweight, is sixteen times as great with the weight in w as with it in W, it follows that with the latter counterbalance a shorter stop can be made than with the former. The distance required to make a stop with the weight in w might be greater than that desired, and the only way to reduce it would be by making w lighter. It is evident that with the counterbalance at W the unbalanced weight of the car can be considerably less than with the weight in w, and that the difference increases with an increase in car speed, and with higher gear ratios. From this it will be evident that when the gear of the elevator is high, the momentum of the counterweight at W has very little effect upon the quick stopping of the car, and on that account the weight of the latter can be more nearly balanced than it can with the independent weight at w. In practice the total counterbalance is put partly in w and partly in W, the object of so doing being to reduce the amount of iron required, and also to add to the safety of the elevator system by providing additional ropes to hold the car, and by taking some of the strain from the car-lifting ropes. Placing the counterbalance weight, or a part of it, at W does not increase the strain on the lifting ropes, as might be supposed on first considering the subject, as this strain is determined wholly by the weight of the car and its load.

HIGH-PRESSURE HYDRAULIC ELEVATORS

The elevators so far considered are of the type known as low pressure, and are actuated with water pressures ranging between about 40 and 175 pounds per square inch. In the early days of hy-

draulic elevators the pressure was obtained by pumping water into a tank located upon the roof of the building, and as a column of water of one square inch cross section weighs about 0.434 of a pound per foot of hight, the pressure seldom exceeded 50 pounds, and in many cases was as low as 30 pounds per square inch. With the advent of high bulidings greater car speed was demanded, and this necessitated increasing the power of the elevator machine. This increase could be obtained in two ways, one by making the cylinder of larger diameter, and the other by increasing the pressure. As space in the buildings was valuable the latter plan was adopted, and in order to obtain the increased pressure, a closed pressure tank was substituted for the open roof tank. The pressure tank had been used years before the time of high-power elevators, in buildings that were so low that the pressure obtained from an open tank on the roof was not high enough to keep the diameter of the cylinder within reasonable bounds; but from this time all elevators installed in high buildings were operated with water drawn from pressure tanks.

The general introduction of the pressure tank resulted in an increase in the working pressure to from 75 to 100 pounds. By this increase the size of the cylinder, the tanks and the piping could be considerably reduced, with a corresponding saving in space and material. With the continued increase in the hight of buildings the pressure has been increased, so that at the present time it is common practice to use pressures up to 180 pounds and even a little higher.

With the view to reducing the size of the cylinders and piping still farther, elevator designers decided to make another increase in pressure, but this time a clear jump was made from 180 to 700 and 800 pounds per square inch, and this higher pressure is now carried on all "high-pressure" systems. The reason why a jump was made instead of increasing the pressure gradually is that the type of apparatus used for the lower pressures is not adapted, in many of its details, to much higher pressures, and since changes in these details were required it was considered advisable to increase the pressure to the highest point that had been found to work with entire success in other fields using similar apparatus, so as to derive the greatest possible benefit obtainable from the use of high pressure.

The differences between high-pressure and low-pressure elevator machines can be pointed out by the aid of Fig. 11, which represents elementarily a high-pressure machine geared two-to-one. A plunger replaces the piston and rods of the low-pressure machine, and the water acts to force it out of the cylinder instead of drawing it in. The advantage of this construction is that only one packing is required, and that is located in the end of the cylinder, where it is easily reached. In the low-pressure cylinder, the piston must be provided with a packing, as well as the piston rods, but as the diameter of the cylinder is generally from 10 to 20 inches, there is ample room for the bolts that tighten up the packing gland, and these can be readily got at when the piston is at the end of the cylinder.

The plunger of a high-pressure machine is finished off true and smooth, but the cylinder is only finished at the end, this part being a casting provided with a pipe connection at the side and a stuffing-box at the outer end. This end casting is bored to fit the plunger. The body of the cylinder is made of extra strong steel piping, the interior diameter of which is considerably larger than the outer diameter of the plunger. As many lengths of pipe are used for the cylinder as may be required, and these are generally connected together by flange couplings.

It will be noticed in Fig. 11 that the cylinder is inverted and that the plunger forces the sheave *B* downward. On this account this type of machine is known as the inverted cylinder. Upright-cylinder machines geared two-to-one have been made, but higher geared machines would be impractical; whereas the inverted machine can be made of any gear desired, and in reality is seldom made less than four-to-one. In a two-to-one inverted-cylinder machine the plunger must weigh considerably less than double the weight of the empty car, otherwise the latter would not be able to run down and lift the plunger up into the cylinder.

If the cylinder were mounted with the packed end up, under the sheave *B*, the weight of the plunger would have to pull up the car when loaded to its maximum

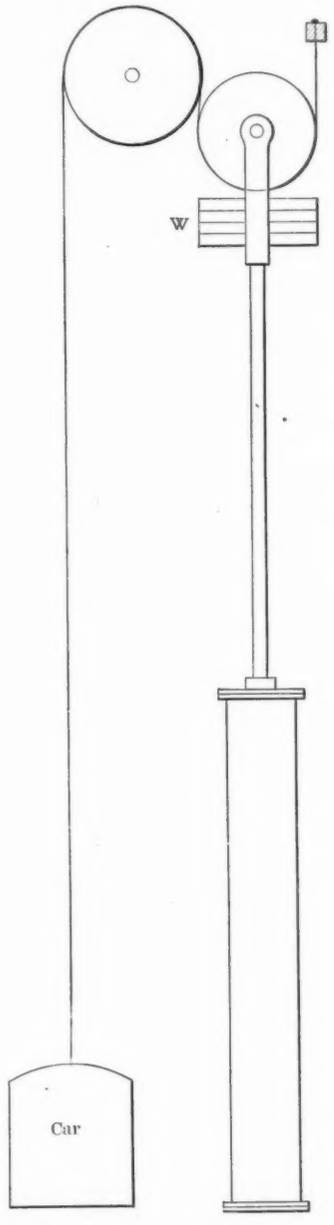


FIG. 9

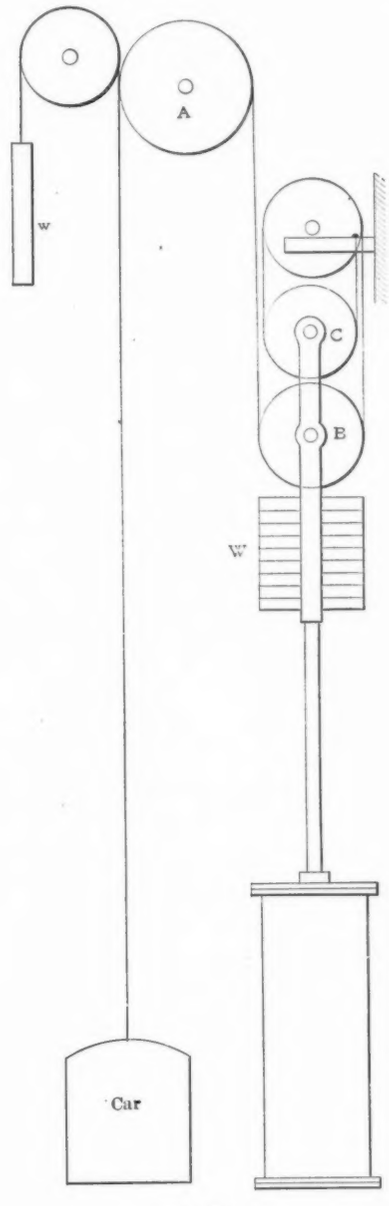


FIG. 10

capacity, hence the plunger would have to weigh considerably more than twice as much as the car with its maximum load. From this it is evident that for the inverted-cylinder machine the weight of the plunger is very much less than for the upright plunger. For any gear greater than two-to-one the upright cylinder would not be desirable on account of the great weight of the plunger necessary to lift the car.

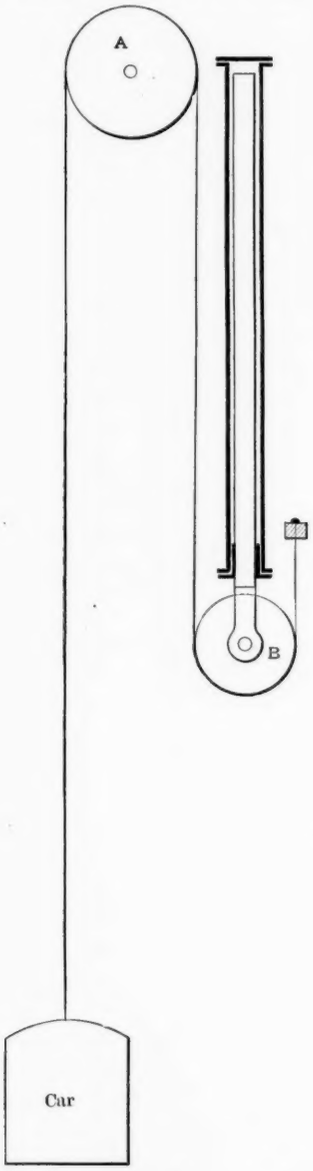


FIG. 11

In the low-pressure system, the water is admitted on top of the piston and forces it down, thereby lifting the car, but in the upright cylinder the water is admitted under the plunger and forces it up, thus permitting the car to run down.

Inverted-cylinder machines are generally geared from four-to-one to eight-to one. A two-to-one machine cannot be used for very high lifts because the plunger could not very well be made sufficiently light. Aside from this difficulty, however, it is cheaper to increase the gear and thus reduce the length of the plunger and cylinder. The general arrangement of an inverted-cylinder elevator geared six-to-one is shown in Fig. 12. As will be noticed, the counterbalance weight is partly at w and partly at W. For medium hight buildings, little weight has to be placed at W as the plunger, sheaves and frame are nearly heavy enough, especially as in such buildings the gear is likely to be not more than four-to-one.

The advantages of the high-pressure system are that considerably less space is required for the machinery and piping, and that the efficiency is higher than that of the low-pressure system. The higher efficiency is due to the fact that the loss of energy in forcing the water through the pipes can be greatly reduced. The energy required to force water through a pipe is made manifest by a reduction, or drop, in the pressure between the point where the water enters and the point where it passes out. The higher the velocity of the water, the greater the drop in pressure, the increase in drop being about as the square of the velocity.

With a low-pressure system operating under about 50 pounds, it might be found to be difficult to keep the loss of pressure between the pressure tank and the lifting cylinder below 20 pounds if the connecting piping were long. This would mean a loss of energy of 40 per cent. If the pressure be doubled, one-half the amount of water will be required, assuming the piston and valve friction to remain the same. As one-half the amount of water is used, the cross-section of the pipes can be reduced to one-half without changing the velocity of the water, and consequently without increasing the loss of pressure above 20 pounds. From this it is evident that the pipe loss can be reduced from 40 per cent. to 20 per cent. by increasing the pressure from 50 to 100 pounds, and at the same time the pipes can be reduced to one-half the cross-section.

If the pressure be raised to 800 pounds, then the plunger area can be reduced to one-eighth, and as one-eighth of the water will be required, the cross-section of the pipes can be reduced to one-eighth without increasing the velocity of the water and, therefore, without increasing the drop in pressure, still assuming constant friction. By increasing the pressure to 800 pounds, and reducing the pipes to one-eighth of the cross-section, the loss of pressure in the pipes is not increased beyond the original 20 pounds, and this is now only 2½ per cent. of the working pressure.

The foregoing is a better showing than could be made in practice because the loss in the pipes and in the piston friction will increase with the pressure and the reduction in diameter, so that the size of the cylinder and pipes cannot be reduced as much as these calculations show, but the sizes would not have to be increased more than one-half, or, roughly, the cross-sections of the plunger and pipes of the 800-pound system would not be more than one-fifth of those of the 100-pound system.

In low-pressure systems the piping varies between about 3½ inches and 7 inches in diameter, and such piping requires considerable space, especially if there are many bends. In the high-pressure system the piping is seldom over 2½ inches in diameter. The pipes required for low-pressure systems are kept down in size somewhat by proportioning them so that when the car runs at the maximum velocity it will only lift about five-eighths of the maximum load, and with the full load the velocity is about one-half of the maximum. With high-pressure elevators, the pipes can be made so as to maintain the maximum car speed with nearly the maximum load, and still be of small size, because the drop of pressure in the pipes is so small a percentage of the total pressure.

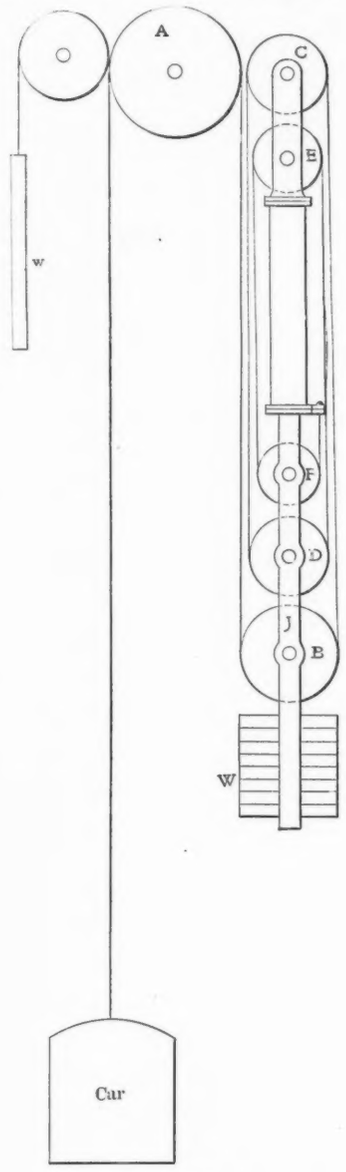


FIG. 12

PLUNGER ELEVATORS

Fig. 13 illustrates in its simplest form what is known as the plunger elevator. Elevators of this type have been made in this country for many years for short rises, but within the last seven or eight years they have come into use for all classes of service and for buildings of any hight. In Europe plunger elevators have been used for many years, but generally for moderate rises—in fact, what we call high rises in this country are practically unknown in Europe, where buildings over one hundred feet high are exceptional. At the present time there are a large number of plunger elevators in operation in New York that rise over 200

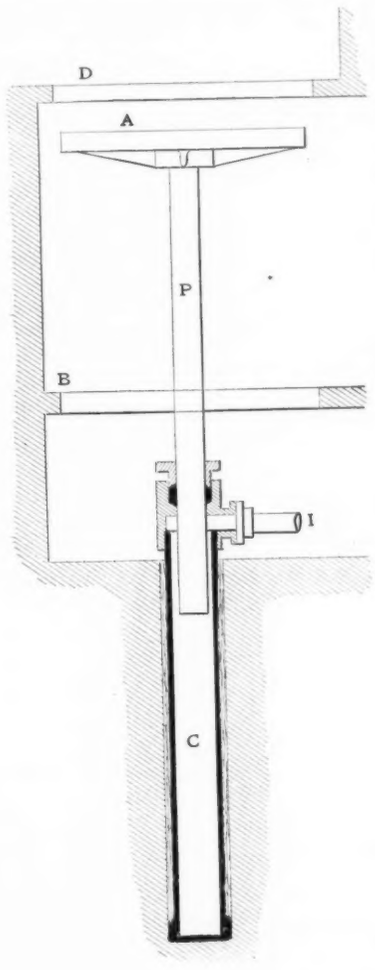


FIG. 13

feet, and in one building they rise about 300 feet. Fig. 13 illustrates what is commonly called a sidewalk elevator, which is arranged to run the platform *A* up from the level of the basement floor *B* to the sidewalk *D*. All this diagram requires to complete it is an operating valve in the pipe *I* and suitable guides for the platform to keep the plunger *P* in line with the cylinder *C*.

The construction of this type of elevator is very simple. The cylinder is made of one or more lengths of steel pipe, the lower end being closed by a suitable cap and the upper end being finished off with a casting bored to take the plunger, and provided with a stuffing-box gland and an inlet at the side for connection with the water-supply pipe. The cylinder is set in a hole in the ground, and is made of a length a few feet greater than the rise of the elevator car. The plunger is made of steel pipe and is turned true and smooth to fit the bore of the top casting of the cylinder. When the rise is greater than one length of the piping used, two or more lengths are joined by means of internal sleeves made long enough to give the joints as much strength as the body of the pipe.

The simple arrangement of Fig. 13 is very satisfactory for short rises, but for any considerable hight it is necessary to counterbalance the car in order to reduce the pressure required to operate the elevator. The plungers are as a rule made of six-inch pipe, which when turned weighs about 16 pounds per foot for standard pipe, and about 25 pounds per foot for extra strong, which is used for very heavy duty. If the rise of the car is 200 feet, and the plunger is made of standard pipe, it will weigh about 3500 pounds, with the end castings. If the car is of equal weight, which is not very far from the average, the total weight will be 7000 pounds, which is a considerable load to lift, especially if the speed is 400 or 500 feet per minute. If a counterbalance of 5000 pounds is provided the power required to operate the elevator will be greatly reduced.

The counterbalance can be arranged in either of two ways, ore of which is illustrated in Fig. 14 and the other in Fig. 15. In the first arrangement the gear between the car and counterweight is one-to-one, and in the second it is two-to-one. Each arrangement has its advantages and disadvantages, and the questions as to which is the better is probably a matter of individual opinion. The plan in Fig. 14 is simple and requires less weight than the two-to-one gear of Fig. 15; but to offset this there is the fact that to be able to make a stop on the up trip within a certain distance, the unbalanced weight of the car must be greater. The reason for this difference is the same as that given in the discussion of Fig. 10.

The counterbalance serves, not only to reduce the amount of power required to operate the elevator, but also to reduce the compression stress that the plunger is subjected to, and thus to prevent the buckling of the latter when stopped suddenly on a downward stroke. Taking the figures for the rise and the weight of car and counterweight previously given it will be evident that if the car is empty, it will fall short 1500 pounds of balancing the counterweight, therefore the latter actually holds up 1500 pounds of the plunger. This means that if the car is empty, the upper end of the plunger is subjected to tension and not to compression. At a point about 90 feet from the top there will be no stress in the plunger, and below this point a compression stress will appear which will increase at the rate of about 16 pounds per foot.

If a load of 1500 pounds is in the car, then the point where there is no stress

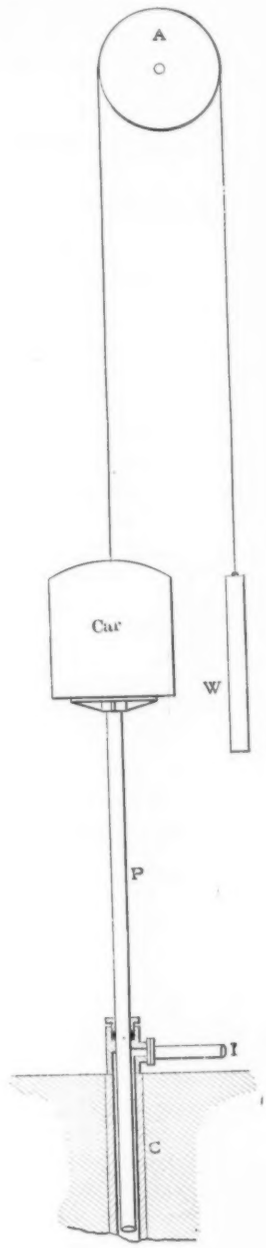


FIG. 14

in the plunger will be at the upper end. With a load of 2500 pounds in the car, which is about the maximum carried in passenger elevators for the average building, the compression stress at the top of the plunger will be 1000 pounds, and will increase below this point at the rate of about 16 pounds per foot, becoming about 2600 at the center point and 4200 pounds at the bottom of the plunger. This is the reason why the plunger, although only 6½ inches in diameter, will not buckle under the load even if made 300 feet long. Another fact that accounts for the rigidity of the plunger is that

only the portion that is in the air, above the top of the cylinder, is liable to buckle, and the compression stress in this decreases as the car rises, owing to the fact that the passengers leave the car at the various floors as the elevator ascends, and when the top of the building is reached, the car is nearly if not entirely empty.

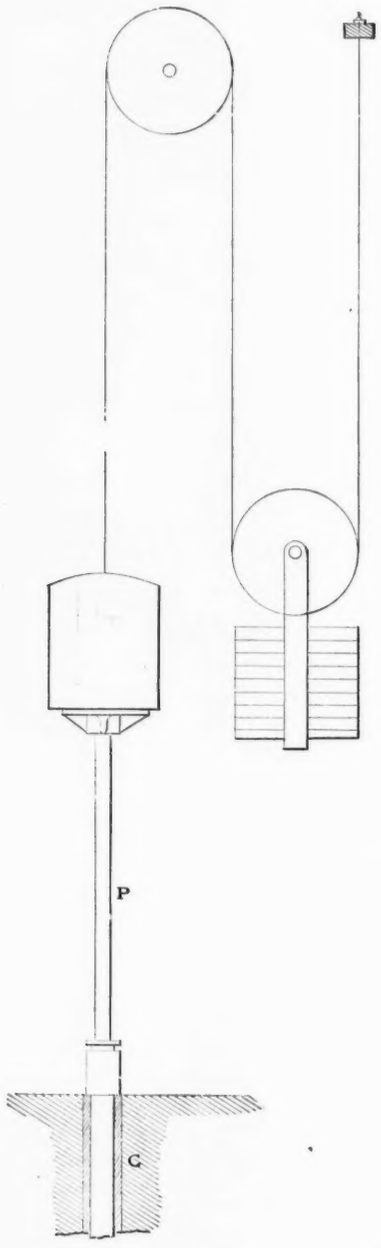


FIG. 15

The compression stress that the plunger is subjected to could be reduced by increasing the counterbalance, but if this were done, the distance required to make a stop would be increased. The unbalanced weight of the car has to be determined with reference to the distance in which stops must be made, and also with reference to the speed of the car. If the stopping distance remains the same, the counterweight must be reduced as the speed is increased, or the weight may remain unchanged if the stopping distance is increased.

The stopping distance is controlled by setting the valve-operating mechanism so that the flow of water cannot be stopped entirely in less than a certain time; while the operator may make a slower stop he cannot make a quicker one. The effect of making too quick a stop on the up trip would be to lift the plunger off the water in the cylinder a few inches; the vacuum produced would assist in bringing the car to a state of rest. On the down trip the effect of too quick a stop would be to put an excessive compression stress in the plunger.

Testing for a Loose Crank-pin

BY W. H. WAKEMAN

A certain engine pounded when the crank-pin passed the centers, and several tests were applied in efforts to locate the cause without success. The engineer was convinced that the crank-pin was loose, but could not readily prove that his idea of the matter was correct. The pin was riveted in place, and where it was beaded on the inside of the disk crank there was no escape of oil or any other evidence to show that the parts did not remain in perfect contact.

The crank was placed on the inside center, the crank-pin boxes were drawn together firmly by the wedge provided for this purpose, and the crank was then turned a short distance by a lever applied to the fly-wheel; but this test failed to turn the pin, although it was carefully marked to determine how much it moved.

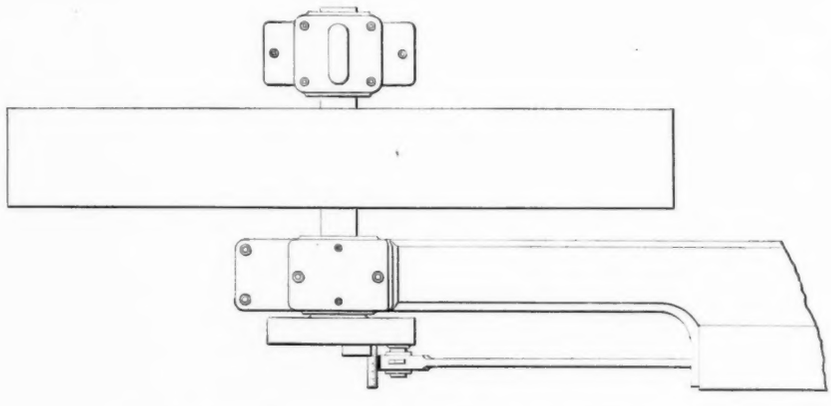
The boxes were again adjusted to working conditions, the crank placed on the inside center, and steam quickly admitted to each end of the cylinder alternately. Although there did not appear to be any unnecessary lost motion while the engine was running, this test showed much more than was expected, making it impossible to tell whether the pin was loose or not.

The main-bearing quarter boxes were then clamped firmly to the shaft by screws provided for this purpose, and the test again applied. Lost motion was less than before, but still too great for a satisfactory test of the pin. Furthermore, the crank, although containing enough cast iron to make it apparently very strong, would spring badly every time steam was applied to the piston, and there was enough lost motion in the crank-pin boxes to prevent a satisfactory test. This was taken up as before, after which a square was held on the face of the crank and against the end of the connecting rod, as shown in the illustration. When steam was again applied to each end of the cylinder, a slight movement of the end of the connecting rod plainly indicated that the pin was loose.

Of course the crank would spring every time pressure was brought to bear on it in either direction, but this movement was not confounded with that of the connecting rod, as only the latter indicates the condition of the pin.

A new pin was roughed out and then turned down nearly to the correct size on a working day. On Sunday the old pin was taken out, which was not a difficult job, as it proved to be loose after the riveted head was cut off, and it had been loose long enough to wear the hole oblong, making it necessary to rebore it in order to secure a perfect fit.

Investigation showed that the old pin had been put in about five years previous, without reboring the hole, although the pin that came with the engine was loose enough to be dangerous. If the job had been thoroughly done at that time, nothing further would have been needed; but in addition to omitting the reboring of the old and worn hole, the new pin was simply driven into a cold crank with a sledge hammer, and the end of it beaded or riveted, which, further experience taught, is not sufficient to make a permanent job. The crank should be heated and shrunk onto the pin, especially where hydraulic apparatus is not available for forcing the pin into place.



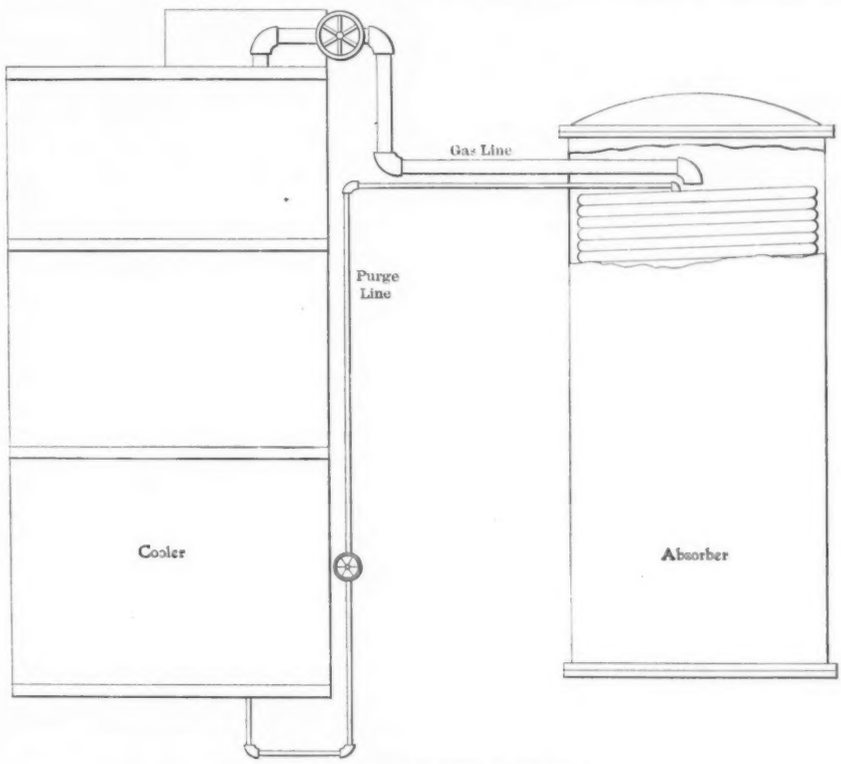
MR. WAKEMAN'S METHOD OF PROVING A CRANK-PIN LOOSE

An Absorption-Machine Wrinkle —How to get Dead Liquor Out of the Cooler

BY W. E. CRANE

While operating an absorption machine it sometimes happens that more liquor gets into the cooler than the absorber will take out, and frequently dead liquor lies in the bottom of the cooler.

The usual manner of getting this into the absorber is to shut off the return gas line from the cooler to the absorber and open a line from the bottom of the cooler to the absorber. This shuts down the machine during the process of pumping out, and the temperature of the brine goes up, often 10 degrees or more, and may require two to four hours. The liquor is evaporated and is taken up in the absorber by the weak liquor. Shutting down a machine for this purpose may really cause a high-temperature brine for several hours.



AN ABSORPTION-MACHINE WRINKLE

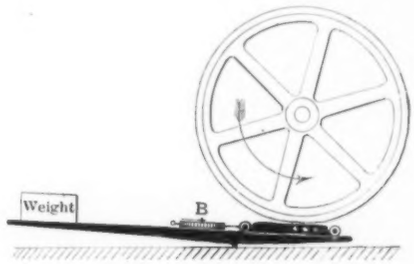
It occurred to me that this liquor in the bottom of the cooler could be gotten into the returning line without stopping the cooler; so I connected the pipe at the bottom of the cooler with the liquor line from the absorber to the pump. In this case it was a one-inch pipe and a gate valve. It is understood that it would not do to attempt to open this valve wide or to attempt to pump it out unless there was a good amount of liquor going from the absorber, as the pump would refuse too rich liquor. The trial was made, and it was found that giving 2 to 2½ turns on this gate valve would clean out the dead liquor, or the accumulation of liquor, at the bottom of the cooler as quickly as by shutting off the gas line, but it will take it out without stopping the operation of the cooler or raising the temperature more than two or three degrees. I have known the temperature of the brine to be reduced during the operation. If there is a large amount of liquor in the cooler, the purge line will work all right for two or three hours, and then the purge line will commence to melt. When this occurs, shut off the purge line for half an hour, then turn it on, and it will start all right again.

As has been stated, the purge line from the bottom of the cooler is connected to the gas line at the top of the cooler, and when both are working together they necessarily interfere with each other.

Where low temperatures are carried, it is necessary to carry the gas line inside the absorber and turn down in the center of the water coils, as, should the gas from the cooler strike against the coils, it would in many cases freeze them. If I were connecting up an absorber, I would not put the purge line into the gas line, but put it into the absorber separately, as shown in the cut. By such an arrangement the two lines would operate independently of each other.

The Sellers Dynamometer

To overcome the difficulties usually attending the measurement of brake horsepower, an ingenious device has been patented by F. Sellers, of 8 Muswell Road, Muswell Hill, N. It consists of a lever, on one end of which is a wooden brake block running on small flanged wheels, and attached to the spring balance *B* in the accompanying sketch. The lever is placed under the pulley, as shown, with the block pressing against the rim of the pulley and the fulcrum resting on the floor. The load is applied by laying any weight on the long end of the lever, and is adjusted by sliding the weight along it,



SELLERS DYNAMOMETER

while the spring balance registers the force applied to the rim of the pulley. The formula is

$$B.H.P. = W \times S \div 33{,}000,$$

where W = the reading on the balance in pounds, and S = the peripheral speed of the pulley in feet per minute.

If the rate of revolutions be the preferred speed factor the formula becomes:

$$W \times D \times r.p.m. \div 126{,}050 = B.H.P.,$$

D being the wheel diameter in inches. The friction of the wheel on the brake block is found to have no appreciable influence on the readings. The idea is certainly clever, and the device will doubtless become very popular.—*The Electrical Review*, London.

The power of small high-speed gasolene engines operating on the four-stroke cycle may be approximated by means of the formula $H.P. = n\,d^2 \div 3$, in which n = the number of cylinders, and d = the diameter of each cylinder in inches. This is based on a piston speed of 800 feet per minute, and a compression of 40 to 50 pounds. For compressions of 60 to 70 pounds, the formula changes to $H.P. = 4\,n\,d^2$.

The Continuous Indicator for Rolling-mill Engine Work

BY ROBERT L. STREETER

The continuous steam-engine indicator has gained a strong foot-hold in this country but recently, probably within the last five years. A few of the larger manufacturers of engines had used it for testing and experimental purposes previous to 1901, but up to that date it had not, so far as the writer has been able to ascertain, been used to any extent on what we might term a purely commercial basis. By this is meant, for instance, measuring the absolute work done by an engine at the same time the steam is measured, so the cost of product turned out by the engine may be computed exactly.

The continuous indicator is no hit-and-miss affair. There is no guessing about it. There are few steam engines doing work where the load is absolutely uniform, and only a small percentage doing work where the load is approximately steady. When one takes a card on the single-card indicator, he trys to get an average diagram. Possibly he hits it, but the chances are against him. Even an average secured from a large number of cards taken at short intervals is unsatisfactory where the load on the engine being indicated may fluctuate from nothing to full load within a few revolutions. The continuous indicator does away with this averaging, but there is another error that may creep in even when the continuous machine is used, and that is in computing the horse-power by using the revolutions per minute. The observations made to ascertain the revolutions per minute are subject to many errors; in fact it is almost impossible to get the absolutely correct speed of an engine at any given time when the engine is subject to sudden variations in speed as is the rolling-mill engine. The continuous indicator, if used properly, furnishes the ideal means for getting around the speed question in rolling-mill engines. First, we will take up a few of the points a good indicator should possess for this work.

1. An outside spring which will allow quick changes to be made. In connection with this, the cylinder cap should be so arranged that it will never need to be removed when the spring is being changed. This will keep the ever present rolling-mill dust from the piston and insure perfectly free motion in the cylinder.

2. The machine should be constructed so it will take a long strip of paper, at least eight feet of it. If the machine is made for a small roll of paper, a long piece may be used if one takes a thin, strong bond paper.

3. The machine should admit of changing the roll of paper without removing the indicator from the engine. This will save many burned fingers resulting from handling a hot indicator. To avoid taking down the indicator to change the paper it is also necessary that the paper drums be arranged so that the end of the new roll may be "threaded" easily. The indicator is often in a place where artificial light must be used, or in a place where all operations are exceedingly difficult so this point is more important than it first appears to be.

4. The drum spring should be strong enough and work freely enough so that there will be no question of the drum getting back into place before the beginning of the next stroke. If the paper pulls a little hard and the spring is not as active as it should be, the drum will appear to go clear back when in reality it does not. The first part of the next stroke, therefore, is used in taking up the consequent slack in the string. This will shorten the cut-off perceptibly if the take-up comes on the exhaust line as it should. Fig. 5 shows two diagrams taken from the same engine under exactly the same conditions. *A* was taken with a single-card indicator and *B* with a continuous indicator in which the action of the drum spring was sluggish. The shortened cut-off in the latter is noticeable. The drum spring of the continuous indicator being stiffer than that of the lighter single-card indicator, the string made for the latter is usually too light for the former. Where two continuous indicators are moved from the same point, a

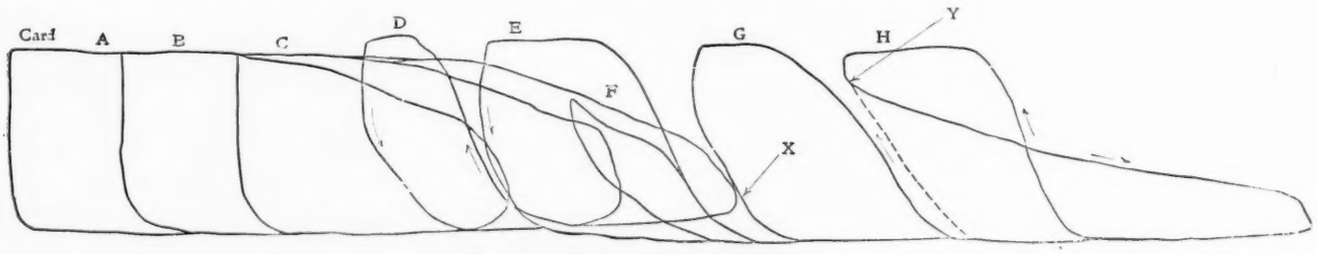


FIG. 1. A SERIES OF REVERSE DIAGRAMS

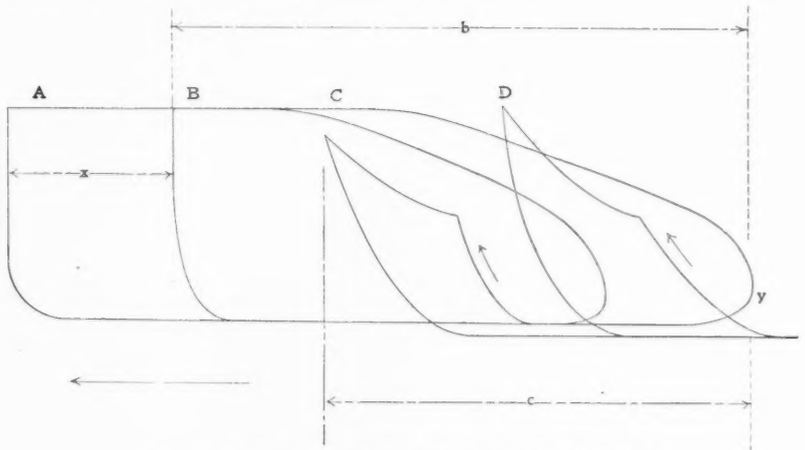


FIG. 2. LENGTH OF WORK AND REVERSE DIAGRAMS

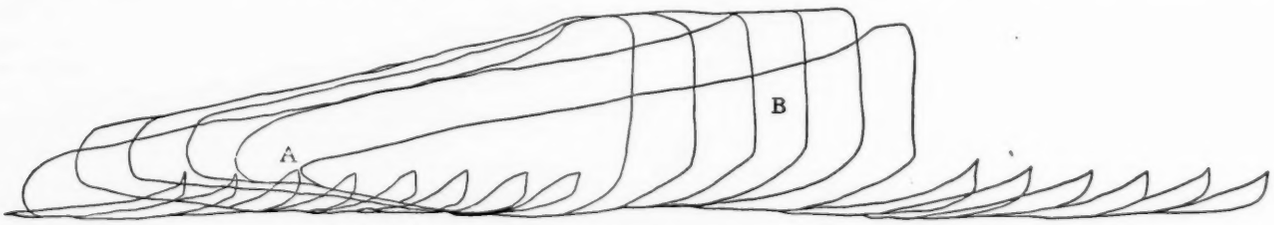


FIG. 4. ONE COMPLETE PASS ON FLY-WHEEL ENGINE ROLLING I-BEAMS

fine wire is very satisfactory in place of a string. The combined tension of the two drum springs is amply sufficient to keep this wire taut.

A lead pencil seems to be more satisfactory for use in the continuous indicator than the brass point working on a prepared paper. Metallic paper is not only more expensive, but is harder to manipulate. The metallic facing on the paper causes greater friction at all points of contact and the force necessary to advance the paper may be greater than the drum spring will exert. A convenient and cheap paper to use is that which is manufactured for some adding machines. This paper has two fine red lines running parallel to the edges, which serve as very convenient guides in working up the cards.

It is desirable to have the take-up of the paper come on the exhaust line of the card and not on the expansion line. There is no positive error if the take-up does come on the expansion line, but it makes the diagram shorter than the stroke of the drum by the amount taken up. This decrease of length is distributed evenly throughout the entire length of the diagram, but it is better to keep the cards as long as possible, especially where the cut-off is not constant. The length of the take-up should be at least three-quarters of an inch. If the cards are too close together it is very hard to separate them, especially with friction and reverse diagrams to be accounted for.

In using the continuous indicator on rolling-mill engines, in order to eliminate the very uncertain factor of speed, it is necessary to take a diagram for every stroke for a given amount of work. There should be an indicator for each end of each cylinder. This is important, not only because the mean effective pressures of the different ends of the same cylinder may vary widely and according to no fixed law, but when the engine is given steam to roll a bar, the steam may enter first in the end of the cylinder where there is no indicator and may leave that end of the cylinder last. This means that on a single-cylinder engine with one indicator we may lose two working diagrams and on a two-cylinder engine with two indicators we may lose three or four working diagrams. On an engine rolling short bars this loss will be a large percentage of the total work. As the bar lengthens and the total number of strokes increases, this percentage will decrease, but it will never be small enough to disregard.

In indicating a reversing engine the diagrams taken during the reversal should be treated as work diagrams and the work of reversing should be charged up against the proper bar and pass rolled. Where short bars are rolled and high speed is attained, this reversal work is a large percentage of the total work, in some cases of blowing or slabbing engines running up as high as 30 per cent. Fig. 1 shows a series of reverse diagrams. At x the valves were thrown with steam in the valve chest. This reverses the direction of the indicator point in traversing the card while the engine is still running in the same direction. At y the direction of the engine changes with steam on. In this case the area shown by the reverse diagram filled out by the dotted line should be charged to the reverse work, and the area formed by the expansion line and the dotted line should be charged to the following pass.

FIG. 3. CLEAN WORKING PASS ON A FINISHING-ROLL ENGINE IN A RAIL MILL

Care must be taken to use the proper length in dividing the area of the reverse diagrams to find the mean hight. Fig. 2 explains the difference in length of the work and reverse diagrams. In that figure, A and B are work diagrams whose lengths are equal to the full stroke of the indicator drum because the take-up comes on the exhaust line. The paper moves in the direction indicated by the feathered arrow. If the valves are thrown at y, the indicator point follows around C in the direction shown by the small arrow, the take-up coming on the line which is now the upper line. This will give a length c for the reverse cards where $c = b - x$, when x is the length of the take-up. If the take-up comes during the expansion stroke of the work diagrams, they will be shortened by the length x and the reverse diagrams will be the full length of the stroke of the drum.

The effect of the inertia of the heavy moving parts of the reversing engine is clearly shown by the continuous indicator. The writer has never had an indicator on a fly-wheel engine when it was reversed, but the effect of large gears is shown in Fig. 1. These diagrams were taken on a 40x60 rolling-mill engine geared to the rolls through a train of five gears giving a ratio of 1 : 4.23. The effect of these gears is very much like that of a fly-wheel. In this case the work of reversing for a given time was equal to about 30 per cent. of the total work. In an engine that was direct connected and doing approximately the same work, the reversing work was cut down to about 10 per cent. of the total.

Fig. 3 shows a clean working pass on a direct-connected two-cylinder reversing engine running finishing rolls in a rail mill. On these rolls there were four passes so arranged that a reversal came only once for each two passes. This cut down the work of reversal to less than 5 per cent. of the total.

Single-cylinder fly-wheel engines are used in rolling mills where there is only one pass, or where the rolls are three high. These engines run continuously and the heavy fly-wheel is an important factor which takes the enormous load the instant a bar enters the rolls. On a 44x66 engine of this type, with a 24-foot 75-ton fly-wheel, running a three-high mill which was rolling 9-inch I-beams from blanks eight feet long, the continuous indicator

showed that the bar made one full pass and half of another before the engine did any work, the fly-wheel taking care of the load. The speed, during this time, dropped from 80 to 60 revolutions per minute, but the engine gradually took up the work here and carried it through the rest of the nine passes, so that by the time the next bar entered the rolls, the speed was back to 80. The friction load on this engine could not be taken with a single-card indicator because most of the diagrams showed no work, or even a negative work. This was due to the governor, which was of the type that regulates the cut-off, being slow. When the continuous indicator was used it was found that the governor would open the valves for two or three revolutions until the fly-wheel attained a speed of five revolutions above the normal of 78 per minute. Then the valves would close and stay closed for from 15 to 20 revolutions, or until the speed dropped down to three or four revolutions below the normal. Then the valves would open again and the whole operation would be repeated.

Fig. 4 shows one pass taken on this 44x66 engine. The bar entered the rolls at A and left at B. On an engine of this type it is almost impossible to separate the work diagram into the different passes of the bar since we may have only three or four series of work diagrams for five or six passes of the bar. But it is just as well to take the total work for the stand of rolls as it is to use the separate passes. The thing we are after, anyway, is the total work per ton of finished product.

In making a test to find the work required in rolling steel rails from ingots, the total work required to roll an ingot into finished rails should be found for a number of different ingots, as the work will vary at least ten per cent. for different ingots. If there are a number of different engines in the mill it will not be practical to have indicators on all of them at the same time. The start should be made at the first engine, and a set of cards taken which shall include several ingots or bars. It is very important that all the passes taken simultaneously should be worked in together. Consequently when one indicator needs a new roll of paper, all should be changed and each roll carefully marked to be used where it belongs. As it is obviously impossible to follow any particular ingot throughout the mill, the next best thing to do is to take each engine in turn, taking a series of cards on each, these to be arranged in order when the cards are worked up.

In integrating these cards it is not necessary to find the mean effective pressure of each diagram, but to find the sums of the pressures for each pass. Then if the sums of the mean effective pressures of the different ends of the cylinders be added together, the two factors $P N$ of the $P L A N$ formula will be the result. Then with the constant $L A$ of each engine as a multiplier, the number of foot-pounds of work for each pass may be found, and the sum of these will, of course, be the total work necessary to roll the rail from the ingot. From the production sheet of the mill may be found the total number of ingots rolled in a given time. Then for the work required to roll a ton of finished rails, we will have

$$\frac{w\,n}{T} = W$$

where w = foot-pounds per ingot
n = number of ingots in a given time.
T = tons of rails in same time.
and W = foot-pounds per ton of finished rails.

The factor n should be the total number of ingots started at the blooming rolls. This will include the work used in rolling cobbles, which is as it should be. Then if the W be divided by 1,980,000, the horse-power-hours per ton will be the result.

The next step is to find the efficiency of the engines with regard to steam consumption. This is best done by measuring the condensed exhaust steam. If a surface condenser be used the matter is simple—weigh the condensed steam for a given period of time and divide this weight by the weight in tons of finished product for the same period of time. This will give the pounds of steam required to roll a ton of product.

When the condenser is of the jet type, it is far more difficult to get the exact weight of steam. The most practical way is to use the condenser as a calorimeter. This necessitates accounting for heat used in the following ways: 1. Heat given up in doing work. 2. Heat given up by radiation. 3. Heat given up to the injection water of the condenser.

Let H_w = Heat used in work per minute.
H_r = Heat used in radiation per minute.
H_c = Heat used in condenser per minute.

$$H_w = \frac{\text{Foot-pounds per min.}}{778}$$

H_r = Radiating surface in square feet $\times$ $(T_s - T_r) \times c$,

where T_s = temperature of steam.
T_r = temperature of atmosphere surrounding radiation surface.
and C = heat units radiated per minute per square foot radiating surface per degree difference in temperature.

The radiation from the exhaust piping, cylinders, receivers, etc., should be computed separately in order to correct for different pressures which determine T_s. The constant C has a value of about 0.04 for bare iron pipes, where the steam pressure is under twenty pounds per square inch.

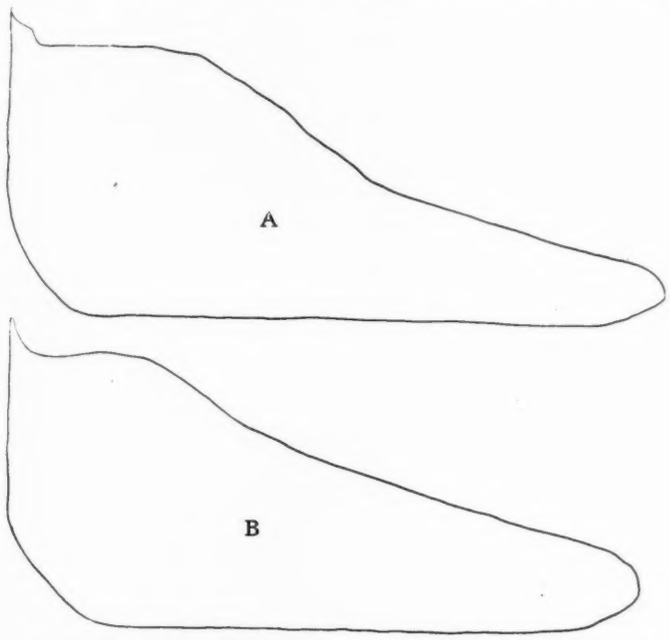


FIG. 5. SIMULTANEOUS DIAGRAMS FROM A SINGLE-CARD (A) AND A DOUBLE-CARD (B) INDICATOR

For the heat given up to the injection water we have

$$H_c = Q_w (T_h - T_c),$$

where Q_w = pounds of jet water.
T_h = temperature of hot well.
and T_c = temperature of jet water.

The jet water should be measured as carefully as possible, as this is the outlet for the greatest amount of heat. Where no meter is available for measuring the water, it may be computed by finding the velocity by careful measurements with the Pitot tube. If the water be pumped into the condenser, the water may be computed by knowing the displacement of the pump, provided the percentage of slip is known.

The total heat being known, the pounds

of steam may be found in the formula:

$$Q_s = \frac{H_w + H_r + H_c}{x\,[H-(T_h-32)] + (1-x)(t-T_h)}$$

where Q_s = pounds of steam per minute.
x = quality of steam.
H = total heat in one pound of steam at boiler pressure.
T_h = temperature of the hot well.
and t = temperature of steam at boiler pressure.

Then for the steam per horse-power hour, we have

$$\frac{Q_s \times 60}{\text{H.P. hrs.}}$$

Rope Drives—Various Methods of Arranging Ropes on the Sheaves

BY R. HOYT

There seems to be considerable difference in opinion regarding the various ways of applying rope to the sheaves in rope driving, viz., multiple- or separate-rope system, continuous-wrap or single-rope system with the rope from one of the grooves running on a traveling take-up device, continuous-wrap or single-rope system with the take-up working directly on all the wraps.

The multiple- or separate-rope system on a horizontal drive where the distance between centers is great enough so the weight of the rope will give the required tension, having the tight or pulling part on the lower side and the sheaves of the same diameter, as in Fig. 1, should be very satisfactory, as old or worn ropes may be replaced by new ones of larger diameter or some of the ropes may be tighter than others and still not alter the efficiency of the drive. It will be noticed in this case that a larger rope does not alter the proportional pitch diameters of the rope on the driving and driven sheaves; but if one of the sheaves is larger than the other, as in Figs. 2 and 3, and a new or larger rope is substituted for a worn or smaller one or if some of the ropes are a great deal tighter than others, a differential action will be produced on the ropes owing to the fact that the larger or slack rope will not go as deeply in its grooves as the smaller or tight one, and consequently the proportionate pitch diameter on the rope on the driver and driven sheave will be changed. The action will depend upon whether the large or small sheave is the driver. If the driver is the largest and of course assuming the slack or large rope is weaker than the combined tight or smaller ones, then it will have less strain on the pulling side; but if the driver is smaller, then the new or large rope will have greater strain on the pulling side. Whether the driver is larger or smaller, a large or slack rope affects the action oppositely to a small or tight rope. Fig. 3 shows how the action is reversed from Fig. 2.

For clearness we will exaggerate the differences in diameter in the sketches and figure the speeds that the different size ropes would produce. We will take A as normal, B 1 inch farther out of the groove producing a difference in diameter of 2 inches; C 1 inch deeper in the groove, producing a difference in diameter of 2 inches In Fig. 1 assume for the normal diameter of driver and driven 40 inches, and 42 inches for B and 38 inches for C, with a speed of 200 revolutions per minute for

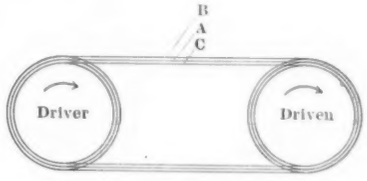


FIG. 1

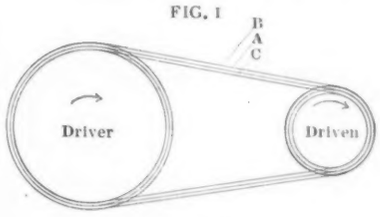


FIG. 2

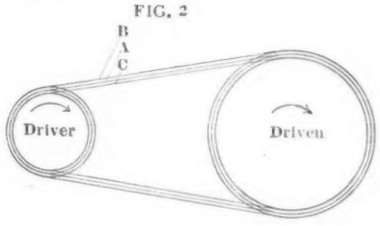


FIG. 3

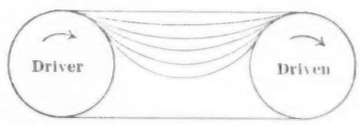


FIG. 4

ROPE DRIVES

the driver. Either A, B or C will give 200 revolutions per minute for the driven sheave, omitting slippage, of course. In Fig. 2, say the normal diameter of the driver for rope A is 60 inches and of the driven 30 inches, a speed of the driver of 200 revolutions per minute will give the driven sheave a speed of 400 revolutions per minute; B, with the driver 62 inches and the driven sheave 32 inches diameter, will give the latter a velocity of 387½ revolutions per minute. With C the driver is 58 inches, the driven 28 inches, and the speed given the latter 414 2/7 revolutions per minute. In Fig. 3 the normal diameter of the driving sheave being 30 inches and the driven 60 inches, a speed of the driver of 200 revolutions per minute will give a speed of the driven member of 100 revolutions per minute. With B, if the driver is 32 and the driven 62 inches, the driven sheave will have a speed of 103 7/31 revolutions per minute; C, with the driver 28 inches and the driven sheave 58 inches, will give the latter a speed of 96 16/29 revolutions per minute. So it will be readily seen what effect a large or small rope would have.

There are some who claim that slack ropes will transmit more power owing to more wrap on the sheaves, while others claim that tight ropes are better. I think if one had a drive and the ropes were all slack and they were troubled with the ropes slipping, the first remedy they would try would be to tighten the ropes. But, if the conditions were like Fig. 3, I don't see that it would be particularly harmful to have some of the ropes longer than others; in fact, it might be well, as the longer ropes would not make a complete circuit as quickly as the shorter ones; consequently the position of the splices would be continually changing. However, it seems more natural, and I should consider it best, to have about the same pull on all the ropes, that is, not have them as shown in Fig. 4 In conclusion for the system, it should be noted that it has no means of tightening the ropes except by re-splicing; it is not as well adapted to various conditions as the other forms; it is the cheapest form to install and in some cases should give excellent satisfaction.

With the continuous-wrap system having the rope from one of the grooves pass over a traveling take-up, the latter has a tendency to produce an unequal strain in the rope. In taking up, or letting out, the rope must either slide around the grooves, or the strands having the greatest pull will wedge themselves deeper in the grooves, producing a smaller pitch diameter than the ones having less pull, making a differential action on the ropes. It is, therefore, probable that it is the differential action that takes up or lets out the ropes, the take-up merely acting in a sense as an automatic adjustable idler. In tightening, when the rope stretches, or dries out, or even in running normal, the greatest pull will be near the take-up, but if the drive is exposed to moisture, and the rope shortens, it will be farthest from the take-up, depending proportionately on the number of grooves the take-up controls; so in large drives it is best to have more than one take-up.

If one should use an unyieldable substance, as, for experiment, a plain wire on two drums wrapped a number of times around and also over a take-up, and the drums were moved together or apart, he would find that the wire would have to slide around the drum; but, of course, with a rope in a groove it is different. The rope will yield some. It will also go deeper in the groove. This system costs

more than the preceding form, owing to extra expense for the traveling take-up, but may be applied readily to different conditions, and will be quite satisfactory in general, if properly designed and installed.

The continuous-wrap system with a take-up or tightener acting directly on all the wraps has practically none of the objectionable features mentioned in the other two forms, and is quick in action, making it applicable where power is suddenly thrown on or off. If the tightener is made automatic, it may be controlled in numerous ways, as with a weight or weight and lever or tackle blocks and weight, etc. It also may be fitted with a cylinder and piston, with a valve to prevent too quick action if power is suddenly thrown off or on. There is ordinarily practically no unequal strain on the rope. This system may be applied to different conditions as readily as the preceding form. Its cost is more than that of either of the others, as the tightener must have as many grooves as there are wraps. It must also have a winder to return the last wrap to the first groove, and to give its highest efficiency it must be properly designed and installed.

In either of the continuous-wrap systems, if a portion of larger rope is used, it will produce a greater strain directly behind the large rope, owing to its traveling around the sheave quicker. In angle work there is always extra wear on the rope in the side of the groove, as only the center or one rope may be accurately lined; so it is not advisable to crowd the centers in angular drives, as the shorter the centers and wider the sheaves the greater the wearing angle. It must be remembered that the foregoing applies to ordinary simple drives as shown in the sketches; where the drive is complicated, it may be necessary to make other allowances.

Good material should be used and competent parties employed for designing and installing rope drives. Don't be hasty to condemn a drive in general because of a poor rope, a flow in a sheave, or because of its being erected in an unworkmanlike manner.

Piping a Steam Pump and an Injector

BY W. H. WAKEMAN

While there is an abundance of suggestions and rules given for piping large plants, there is but little attention paid to the arrangement of pipes and valves in small plants, although there are many more of them than of large ones, consequently we often find piping done as in Fig. 1.

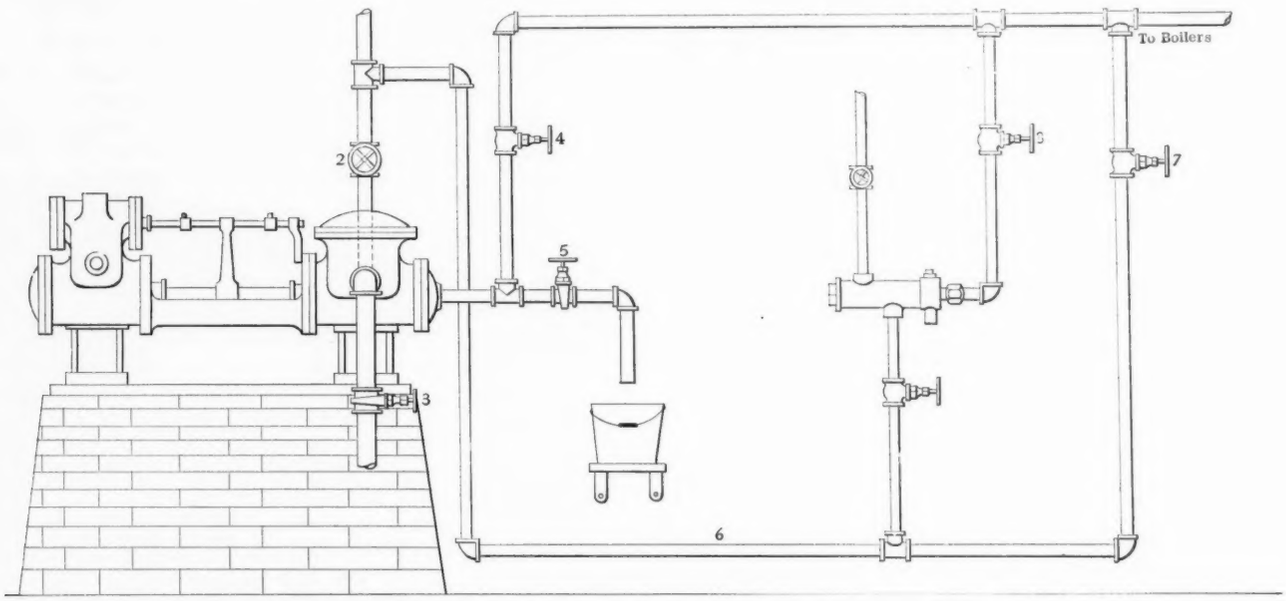


FIG. 1. THE ARRANGEMENT TO WHICH MR. WAKEMAN FINDS OBJECTIONS

The valve 2 admits water from the street main under pressure to the pump. When taking water in this way, the gate valve 3 is closed, as it controls a suction pipe for taking water out of driven wells. When well water is wanted, 2 is closed and 3 opened; but it is a gate-valve which it requires about nine revolutions of the wheel to open, assuming it to be a 2-inch valve, and it is always difficult to repair a gate-valve when out of order.

A globe valve in the discharge pipe is shown at 4, which is objectionable because it creates friction when used on a line of water pipe. The gate-valve 5 is used for drawing off water for various purposes. Just imagine the inconveniences of opening a gate-valve every time water is wanted.

The pipe 6 conveys water under pressure from the street main to the feed-pipe line for the purpose of filling the boilers after they have been cleaned and inspected. There is a globe valve 7 in it which, of course, retards the flow of water more than is necessary. Cleaning boilers is always a disagreeable job that usually must be done on Sunday or a holiday, therefore every effort should be made to shorten it as much as practicable, and getting the boiler filled quickly is an important feature. In one case to which my attention was called some time ago, it required about five hours to fill a seventy-five-horse-power tubular boiler because proper methods for installing the water piping had been ignored.

Anything that increases friction reduces the amount delivered, yet a globe valve was put in the delivery pipe of this injector at 8, although the machine was none too large for the service required.

Fig. 2 illustrates the same pump and injector, with improvements made in the piping and valves. No change is shown at the globe valve 2, because the objection to a gate-valve is about equal to the advantage gained by its use, for although the globe valve 2 increases friction in the water system, the pump always gets a full supply, and a globe valve with a soft rubber disk can be easily repaired by putting in a new disk, or by repairing the seat with a valve file when it is required.

The suction pipe for drawing from the wells is now fitted with an angle-valve 3. This creates no more friction than an ordinary ell, and it can be opened with one-half of the revolutions of wheel required by a gate-valve. It can be repaired as easily as a globe valve, and less joints in the piping are required.

The delivery pipe is now fitted with an angle-valve, 4, and water for various purposes is drawn through another angle-valve, 5, as shown. There is positively no obstruction to the free flow of water to fill empty boilers, so far as valves are concerned, through the pipe, 6, as it is

controlled by the cross-valve 7 that causes no more friction than an ordinary tee.

Delivery of hot water from the injector is controlled by the cross-valve 8, giving all of the advantages, so far as quick opening and ease with which repairs can be made, gained by use of a globe valve. The suction pipe of the injector is still fitted with a globe valve, because it can be adjusted to meet various demands more easily than a gate-valve, and when opened wide it gives more water than the injector can take.

Practical Troubles with Steam Turbines

BY E. RUSSELL

In the operation of steam turbines frequent cases of stripping the blades occur, most frequently in turbines of the Parsons type, on account of its necessarily close clearances. Some of these cases are solvable, some are guessed at, and others remain a mystery. The chief causes of these strips are bad workmanship, want of balance in the spindle, warping of the cylinders, and not infrequently foreign substances coming from dirty steam pipes and passing the strainer.

I have had one or two bad strips caused through the breaking of the stop pieces which are let into the cylinder, one at each end of a half row of blades where the joint between the top and bottom halves occurs. These are slightly dovetailed and driven in firmly, and often in the driving would get bent a little, in which case they would be straightened, with the result that in six cases out of ten they would be fractured. Consequently, on the blade coming into the slightest contact with the rotor, it would drop off, and cause in most cases the stripping of three or four rows, or perhaps a complete stage. This is eliminated in the more recent machines by using wrought-metal stop pieces.

Some people appear to think that if one row of blades gets stripped, the whole lot must go; this is quite wrong, as any turbine man knows; for in cases of small strips (three or four rows), the accident is often not discovered until the machine is being shut down, when a slight ringing sound may be heard just as the stop valve is shut, and the stripped blades, which have been held up until that time against the first uninjured row of cylinder or guide blades, are falling down to the bottom of the cylinder, where they get caught by the spindle and are ground into pellets, which turbine people are, I am sorry to say, quite used to seeing.

Again in calking in the packing pieces, one or two may get missed altogether, in the very small blades, or improperly calked in the larger sizes; but this is not so serious a matter as it might appear at first, especially if the packings are of yellow metal, as the expansion of the rest of the ring including the blades will wedge it in quite securely. At the same time it is not a good thing to have any loose.

In most modern machines the packings are of annealed steel, and the blades of delta metal. The advantage gained by using steel packers is that these, being very nearly of the same composition as the rotor, expand very nearly the same, and there is no tendency to force the blades out of the grooves, which there is a slight danger of in the case of yellow-metal packing and blades in a steel drum.

Quite a few strips are accounted for by the old-fashioned method of lacing a strip of delta wire laid in a notch near the end of the blade with fine copper wire, and then brazing these, with the result that unless the man was an expert, there would be danger of burning the blades, and, should they touch, a strip was very likely to follow. Another danger from this method lay in pieces of spelter from the brazing sticking to a blade under the binding wire, and getting detached when in service.

Of all binding strips, the best that I have seen is that known as the "comma lacing." This consists in punching near the outer ends of the blades and in the center of them a hole of a section like a keyhole. Through this is passed a delta-metal wire of the same section and the tail of this is bent over at right angles between the blades with a special pair of pliers, the jaws of which are of the correct width for the space between the blades. There is no brazing in this method, and it makes a good, strong, quick job.

Of course these lacings are quite different from Willan & Robinson's shroudings, which is a great improvement on all previous blading arrangements, as in the first place the blades are carried in a mechanically spaced half ring on the root, and the shrouding on the tips is also mechanically spaced and riveted up so that in blading the complete ring of blades is put on in halves. In fact, with their method of blading it is possible to reblade a large machine in forty-eight hours against probably ten times this time with the old method of blades and packings and their necessary bindings and lacings.

Again a deal of trouble is derived from improperly designed cylinders, which, on heating up to a temperature of say 550 degrees Fahrenheit at the steam inlet, which is good practice with turbine work, will warp, causing the blades to rub on their tips, rotor blades on the cylinder walls and cylinder blades on the rotor, with the result of a strip if not stopped in time.

In nearly every case it will be found

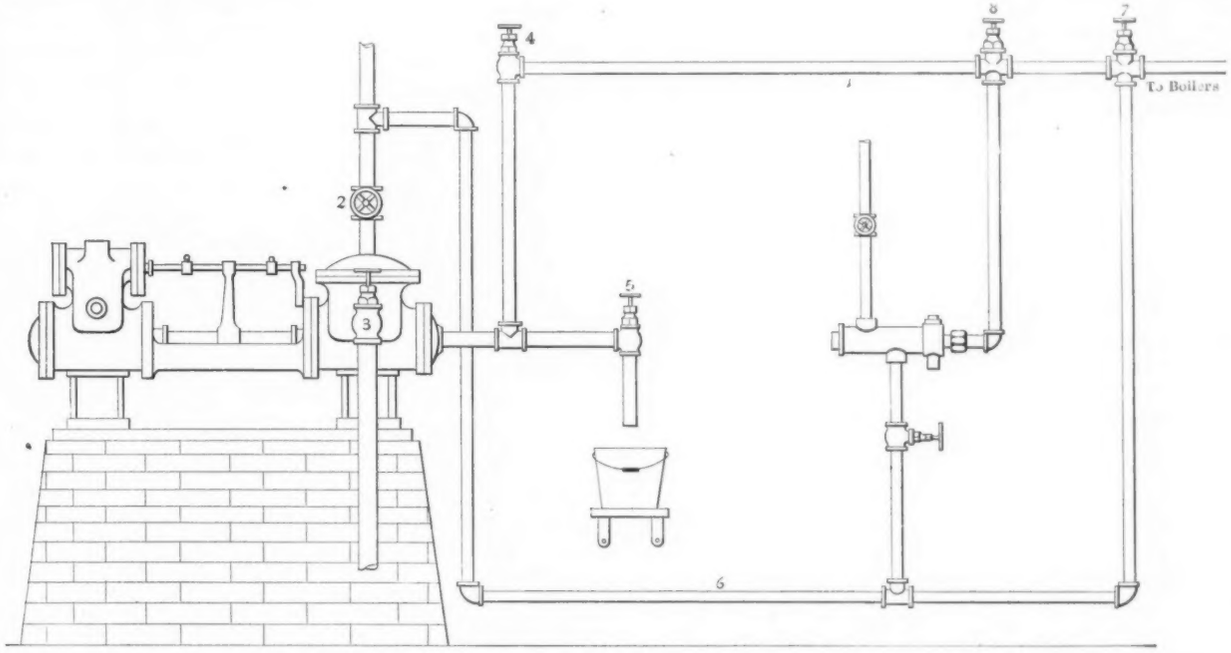


FIG. 2. THE SAME WITH ANGLE- AND CROSS-VALVES SUBSTITUTED

that turbine cylinders become slightly elliptical after a little time in service, the largest measurement being vertical. Of course this is easily accounted for by the two flanges running around where the cover and bottom halves are bolted together, which forms a very rigid band around the machine. In the best practice turbine cylinders are steamed for a considerable time before the boring is finished, as this seems to bring up any bad sets that are likely to take place, and allows them to be taken out with the finishing cut.

Again, when the average salesman calls to see a likely buyer, if the buyer is not an engineer he would imagine that all he required to do when his turbine was installed would be to start it up, and from that date all repairs and trouble would cease, which, so far as repairs go, has a fair amount of warrant in fact; but he forgets that there may be such things as impurities in the water which will form a deposit on the blades and in some cases make it necessary to have the spindle out once or twice a year, and that it may take anywhere from a few days to a week to clean it out, according to the size of the machine, and the nature and quantity of the deposit.

At first it appears strange that steam, passing through a turbine at 500 to 700 feet per second, should allow any deposit to remain on the blades; but the steam has little weight, whereas the deposit, although in minute particles, has considerable, and therefore on being carried forward by the steam instead of flowing off the guide or moving blade, really passes through the stream of steam which is carrying it and strikes the blade, where if it is at all moist it sticks, and is gradually added to until the power of the machine is considerably reduced. In one instance a 5500-kw. machine got into such a state from this cause that it would only take about 60 per cent. of its load. Another peculiar thing in this machine was the effect upon some of the low-pressure blades. The last three rows had been left out for some reason when the turbine was erected, to be put in later. The low-pressure blades became more or less choked with deposit, in some cases for the full length of the blade, which meant that more steam had to pass through the spaces left open, with the result that upon opening the machine, it was found that some of the blades had been cut away altogether by the action of the increased velocity of the steam, and others were unrecognizable as blades. In the London district where this occurred, a great deal of trouble has been caused with this deposit, which is showing up very heavily in nearly all the turbine plants; of course more heavily in wet-steam plants than superheated ones. In the latter the deposit generally occurs heavily in the last two stages, the last two rows as a rule being very bad.

A good method of cleaning this deposit off is to take the spindle out, and spray it well with paraffin under a little pressure. Let this soak in a little, and most, if not all, of the deposit will be washed out with a strong stream of water.

This deposit does not seem to occur so badly in impulse machines, and most of them are pretty free from strips. The chief trouble is due to the blades coming into side contact, as the side clearance has to be kept very small to run efficiently in this class of machine, although occasionally a de Laval will shed a blade due to centrifugal force, the rim speed used in this type being very high.

Fitting the High-pressure Cylinder of an American Ball Engine with a Special Metallic Packing

BY WM. KAVANAGH

Some time ago it was decided to equip one of the compound Ball engines in our plant with the latest metallic piston-rod packing. One of the Corliss engines in the same plant had been fitted with this packing and its success on the Corliss resulted in the decision to try it on the Ball engine. The construction of the Corliss engine is well adapted to the application of this packing since the stuffing-box is easily accessible, it being only necessary to remove the crosshead and gland to put the packing in the stuffing-box. With an engine of the American Ball duplex type it is very different, as will be readily understood by those familiar with the engine. In fact, the packing of stuffing-boxes on all engines of the inclosed type is difficult, especially with the metallic packing about to be described.

The accompanying sketch shows a longitudinal section of the packing under consideration. The packing proper is divided into two sections of six rings each, the rings in each section being numbered from 1 to 6 in the sketch. The first five of these are of babbitt and the sixth of brass; all six are split into semicircles and inclosed in an iron cage, *C*. Beginning at the left of the entire group of packing rings and other members, there comes first a brass ring *N*, of the cross-section shown in the sketch; next a flat brass ring *I*, then the cage *C* containing six packing rings; against ring No. 6 is another flat brass ring *I'* which is pressed against the packing ring by the helical spring housed in the cage *B*. This cage is shouldered at the right-hand end so that the spring reacts against the cylinder head. Its right-hand end is also hollowed out to take another brass ring like the one (*N*) first mentioned, and this is followed up by an arrangement of parts which is a duplicate of that just described. The inner end of the spring cage *B'* rests against the bottom of the original stuffing-box in the cylinder head.

It will be obvious from the sketch that the pressure of the two helical springs causes the packing rings *1* and *2* of both groups to hug the piston rod, these rings being tapered on the outside, housed in the tapering part of the bore of the cages *C*, and split across their centers. The left-hand half of the complete packing is contained in a large housing *A*, which is bolted against the cylinder head with a gasket ring *D* interposed to prevent the escape of any steam that might leak past the right-hand half of the rod packing. The over-all length of the complete construction, with the springs extended, is 13½ inches; this must be reduced, by compressing the springs, to 11½ inches when the packing and housing *A* are in position, and it was this fact that made the operation of putting the packing in place so awkward.

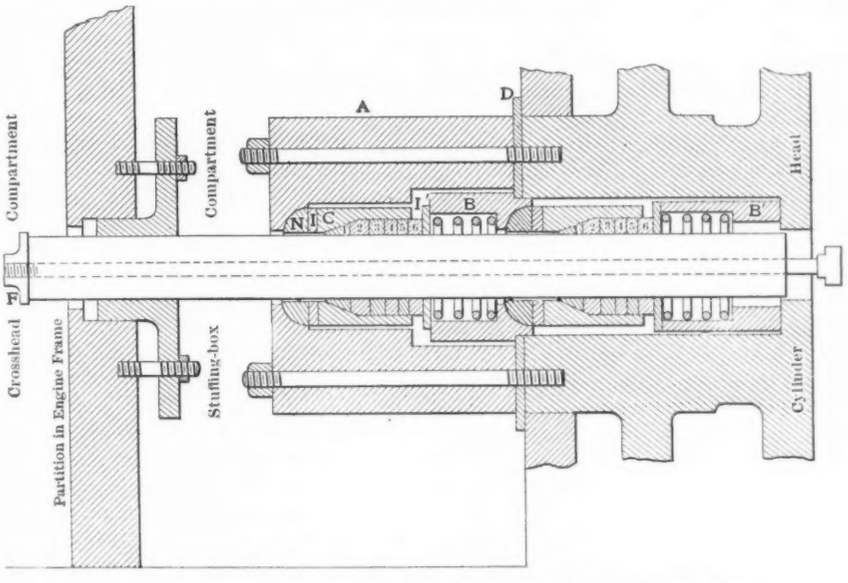
The compound engine to which this packing was fitted, as most POWER readers doubtless know, has its low-pressure cylinder located above the high-pressure, both cylinders being horizontal and in the same vertical plane. A door is provided in the frame of the engine to permit access to the lower stuffing-box for the purpose of packing or tightening the gland; this door is sufficiently large to permit inspection and cleaning of the crosshead compartment, from which the piping system for distribution of oil to the different bearings leads out. This door also affords access to the lock-nut on the crosshead end of the piston-rod.

A partition separates the crosshead compartment from the stuffing-box compartment, as shown in the sketch and its distance from the cylinder head determines the amount of room in which the packing must be done or repairs made. This distance in the engine under discussion is about 7 inches. The object of the partition is to prevent any steam that might escape through the main stuffing-box from getting into the crosshead compartment, where it would condense into water, which would then find its way to the oil circulating pump, and interfere with the oil circulation. To prevent this the partition is fitted with a stuffing-box and gland. While it is possible to reach down to the nuts of the lower main stuffing-box and tighten them by feeling, the stud-bolts and nuts are more easily reached through the door in the side of the frame. In any case only one hand can be used to work within this compartment. This description of the partition and stuffing-box compartment is necessary in order that the reader may appreciate the difficulty of applying under consideration.

In order to insert the packing the rear cylinder head was removed and the crosshead was moved into position so that the lock-nut on the piston-rod was accessible: the positions of the crosshead and piston were noted before the piston-rod was withdrawn. Then the lock-nut was loosened and the piston and piston-rod withdrawn from the cylinder. The glands of

the cylinder and partition stuffing-boxes were then removed and all of the old packing taken out. A wooden rod was substituted for the real piston rod, and on this the packing was fitted and pushed into position. A hole sufficiently large to receive a ½-inch iron pipe was bored lengthwise through the wooden rod.

The cages, springs, housing, gasket and stud-bolts were lowered into the stuffing-box compartment, the wooden rod was pushed through the complete construction and the partition, and a piece of iron pipe about four feet long, with a tee on one end of it, was pushed through the hole in the wooden rod. A pipe flange was then screwed on the inner end of the pipe, as shown, this flange having been previously turned down to pass through the stuffing-box in the partition. A small tackle was attached to the tee on the other end of the ½-inch pipe and made ready to pull on the pipe. One man reached down with one hand between the partition and the cylinder head, while another worked through the door in the side of the frame. When they got the packing into position, the wooden rod was pulled through from the rear of the cylinder until the ½-inch flange brought up against the housing *A;* then the tackle was used to compress the springs and permit the studs to be set into their holes and the nuts tightened down. Before the packing was lowered into the compartment, the positions of the stud-bolt holes in the cylinder head were marked by means of white paint, so that some idea of their location could be obtained after the gland was lowered and pulled into place. After the packing was fixed in position it was only necessary to remove the piece of pipe and wooden rod and replace the piston, piston-rod and cylinder head.



THE SPECIAL PACKING IN POSITION ON THE SUBSTITUTE ROD

One feature of the operation that should be mentioned is that the threaded end of the piston-rod caused considerable difficulty in passing it through the packing. In order to avoid this it is advisable to fill up the threads on the end of the rod with a hard wax or soft metal before pushing the rod through the packing; of course, after the rod is through the wax or metal can be removed.

Three attempts had been made at this job before the writer undertook it, and each attempt failed, owing to the fact that while the packing could be readily lowered into position there was no means by which the springs could be compressed, so as to permit the stud-bolts to be set into position.

Motor and Dynamo Troubles

BY W. T. FERNANDEZ

When a new motor exhibits a tendency to develop heat in one bearing, the bearing should of course, be examined. If it is found in good condition, the cause of the heating will probably be found in the thrust of the armature shaft against the bearing, which may result from either of two conditions. First: the machine may not be level and the armature shaft may dip toward one end. Second: the magnetic centers of the pole-pieces and armature may not be in line, that is, the pole-pieces may not be exactly centered in their relation to the magnetic center of the armature axially, and as the tendency of the armature is to run to the true magnetic center, it will automatically tend to adjust itself to that position, which may cause the shaft collar to rub against the bearing at one end and cause heating of that bearing.

The unbalanced magnetic condition mentioned may have been caused by forcing the armature core not quite far enough or a trifle too far on the shaft, in the factory. Some motors are furnished with slots in the field-magnet yoke through which the magnet cores are bolted to the yoke, and the cores may be shifted a trifle to the right or left to compensate for any slight axial unbalancing of the magnetic center as compared with that of the armature.

Fig. 1 illustrates the condition mentioned. It will be evident that if the

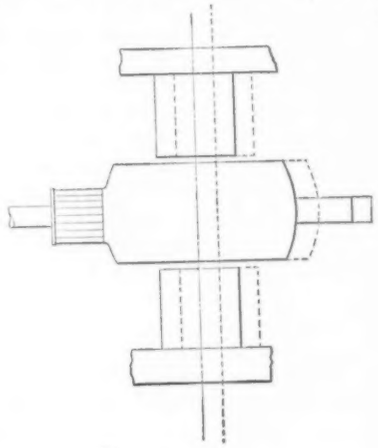


FIG. 1

magnet cores are shifted to the positions indicated by the dotted lines, the armature will naturally follow the new centers of the poles and run clear of the left-hand bearing. Moving the magnet cores 1/16 of an inch will, as a rule, be sufficient to give relief in cases due to unsymmetrical magnetic centers. If the field-magnet yoke is not slotted, a light cut may be taken off the shoulder of the shaft at the end which rubs against the bearing, in order to obtain the desired clearance.

Sparking at the brushes of a new motor may be caused either by improper fitting of the brushes or roughness of the commutator; it is usually due to the former The brushes should be sandpapered to a true fit to the commutator by drawing a slip of sandpaper, the width of the brush, back and forth under it, care being used to move the sandpaper on true line with the curve of the commutator, until all of the face of the brush bears evenly against the commutator. Thus started, the brushes will maintain a true bearing on the commutator.

Commutator lubricant should be used sparingly, as an excess of oil or other lubricants tends to form a gum around the face of the brushes, causing pitting to take place, and this will become a prolific cause of annoyance. Paraffin touched to the commutator once every few hours after the commutator has become warmed up is an excellent lubricant for the purpose and does not form any pitting points in the brush faces.

The connections of a motor should be tested once a week and the machine thoroughly cleaned at the same time. If the machine is equipped with self-oiling bearings, the oil should be changed once every three months and the bearings

washed out with kerosene; that is cheaper than renewing a burned-out babbitt sleeve and insures continuous service.

If a flash appears under each set of brushes as a certain commutator bar passes under each set, it may be assumed that one armature coil is open-circuited, which condition may be repaired in a temporary manner by "jumping" the damaged coil as shown in Fig. 2; bearing in mind that the "jumper" wire must be at least as large and of as great a carrying capacity as the armature wires.

If on starting a motor for the first time, the armature moves in jumps and considerable flashing appears at the brushes, assuming that they have been set at the neutral point of commutation, the polarity of each pole-piece should be tested with a compass to make sure that the alternate poles are of the same polarity, as a reversed winding on one pole will cause two adjoining poles to be of the same polarity and frequently produce the condition mentioned; reversing the connections to the field coil involved will remedy the trouble.

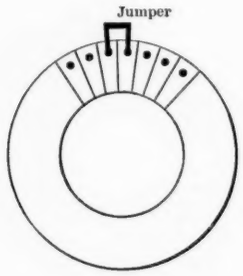


FIG. 2

It is a good practice to have the frames of all low-tension motors clear of any connection with the ground and to test both armatures and field coils for a "ground" at frequent intervals, which may be done as follows: Connect the frame of the machine to be tested to the ground—a water pipe if possible—and then connect one lead of a voltmeter or lamp to a source of current supply and the other lead to the armature and field connections of the motor under test. The voltmeter will indicate the amount of leakage to ground, as will the lamp, roughly, but of course the lamp will not indicate any leak much under forty volts. It is perfectly safe, however, to operate any machine on which there may be a ground, if the ground will not light a 16-c-p. 110-volt lamp. This applies to apparatus operating on three-wire grounded-neutral systems only as a ground on apparatus operating on a two-wire system will not cause any damage until a second ground is formed, but the same test may be applied to such apparatus by disconnecting the armature and field leads of one terminal block on the machine and then testing from that terminal block to the frame of the machine with the switch of other lead closed on the switchboard; the results, if windings are grounded, will be the same as in the first test.

As motors are so connected that the field circuit is always excited in conjunction with the armature circuit from an external source of supply, they are free from a trouble sometimes met in starting self-exciting shunt-wound dynamos, namely, a "killed" field magnet, which prevents the machine from building up its voltage. That condition may be brought about by a severe short-circuit on the system on which the machine has been operating or, as has sometimes happened, from no apparent cause. Cases have been known where machines have been shut down in the regular manner and when started again a few hours later, have failed to build up any voltage. The remedy is simple and consists of disconnecting either the negative or positive field terminal from the machine and connecting it through a small lamp bank to a source of current supply of the same polarity, and connecting the other field terminal to the opposite supply polarity by any suitable means. After the current has been permitted to pass through the lamp bank and field coils for a few minutes, the field terminals should be restored to their respective terminal blocks and the machine brought up to speed, when it will be found all right.

The brushes of a dynamo should always be set at the neutral point when the machine is started and carried forward—that is, in the direction of rotation—as the load is taken on and carried backward, or against the direction of rotation, as the load drops off.

Sandpaper sprinkled with a few drops of oil and applied for a few minutes each time the machine is shut down will maintain a commutator in excellent condition for a long time and save the expense of frequent turning-off.

It should be obvious that all parts of machines must be kept as clean as possible and moving windings free of oil. Where possible, the armature should be blown out by compressed air at about 50 pounds pressure per square inch, but where that is not possible, a powerful bellows may be substituted, to remove the particles of copper and carbon which gather in the crevices of the windings.

A Large Furnace Gas-Driven Electrical Equipment

At Gary, Ind., there is being constructed by the United States Steel Corporation an enormous plant for the manufacture of steel, which when completed will make use of substantially five million tons of ore per year and will produce annually nearly two and one-half million tons of steel. Along with the completion of the various buildings of the new plant an entirely new city is going up, with parks, theaters, streets and water and sewage systems all planned.

Quite in keeping with the immensity of this project are the plans for the detailed equipment. Practically all the machinery in the steel plant will be driven electrically, the gases from the blast furnaces being used as far as possible for generation of the necessary power. One of the first buildings to be equipped will be the rail mill, to be driven by induction motors which are not only of interest because of their size but because of the special method of control. Moreover, this will be the first steel mill in this country to operate its rolls by induction motors. Six General Electric reversible three-phase induction motors ranging in capacity from 2000 to 6000 horse-power will drive the main rolls. Because of the heavy and sudden overloads to which these machines are necessarily subjected, each motor is provided with a heavy fly-wheel and has an overload capacity of 50 per cent. for one hour.

The system of control was designed especially for this service. A master controller is used from which the main-line oil switch can be opened or closed, the reversing switch thrown in either position, and the resistance cut in and out of the rotor circuit in successive steps by means of electrically operated switches or contractors. The main-line switch is a triple-pole single-throw electrically operated oil switch and the current passing through this is led through the reversing switch, a double-pole double-throw electrically operated oil switch. The system is fool-proof. It is impossible to operate the reversing switch unless the main-line switch has been previously opened, and if the main-line switch has been opened by the overload trip, it cannot be closed without first bringing the controller to the "off" position.

In order to assist the fly-wheel in delivering its stored energy to the motor during overloads, "slip relays" are provided. The current of the motor passes through the actuating coil of the relay and whenever the motor is subjected to an overload, the relay operates to cut in a small portion of the rotor resistance in each phase, causing the motor to slow down gradually and the fly-wheel to give up a portion of its stored energy. When the overload is relieved the reverse operation takes place and the motor speeds up, storing more energy in the fly-wheel.

Mr. Junge says that the most economical way of reducing heat losses through the exhaust is by utilizing the same for raising steam in an exhaust boiler. As high as 160 pounds per square inch can in this way be generated, and used for heating and other purposes. Generally 10 per cent. or more of the total output can be recovered from the exhaust of gas engines.

ANNUAL MEETING OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Interesting Papers on Gas Power—Steam Turbine Characteristics—A New Transmission Dynamometer—Engineering Card Index System—An Improved Boiler Setting—Test of a Rotary Pump

Prof. F. R. Hutton, the New President; Calvin W. Rice, Secretary

The fall meeting of the American Society of Mechanical Engineers, held at New York during the first week in December, was the largest in the history of the society, the registration numbering up to 1352. Disappointed in its expectation to hold the meeting in the new Engineering building, the society accepted again the hospitality of the New York Edison Company, and the meetings were held in the Edison building on Twenty-seventh street.

A MECHANICAL ENGINEERING INDEX

Professors W. W. Bird and A. L. Smith describe an index of information on mechanical engineering which has been in use for some three years at the Worcester Polytechnic Institute.

In general the index consists of a series of tab cards on which are printed the names of the various subjects connected with mechanical engineering. All of these cards are arranged in alphabetical order and are easily found. Starting with the card Mechanical Engine, one finds upon it

See Also Construction,
Machine Design,
Materials of Construction,
Power Transportation.

Turning to the "Power" tab card he finds

See Power Consumption,
Power Cost,
Power Measurements,
Power Plants,
Power Transmission,

and following out one of these divisions, as, for example, Power Transmission, he finds a whole line of topics which are still further subdivided.

The tab card may be simply a directory card with the directions for finding the required information. If, however, the tab card says "See Also", then the references are simply suggestive, while back of the tab card itself are the regular index cards. These are of various colors and have the following significance:

Blue—books.
Salmon—catalogs.
Green—theses.
White—transactions and proceedings.
Amber—periodicals.
Red—lantern slides.

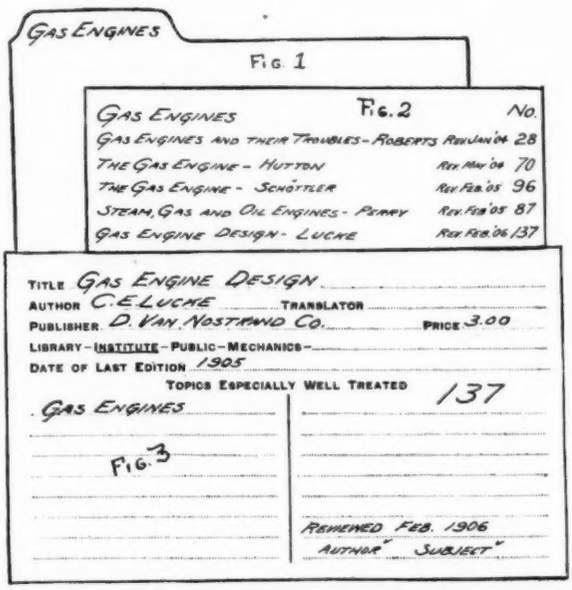

GAS ENGINES
FIG. 1
GAS ENGINES FIG. 2 No.
GAS ENGINES AND THEIR TROUBLES - ROBERTS Rev Jun '04 28
THE GAS ENGINE - HUTTON Rev Mar '05 70
THE GAS ENGINE - SCHÖTTLER Rev Feb '05 96
STEAM, GAS AND OIL ENGINES - PERRY Rev Feb '05 87
GAS ENGINE DESIGN - LUCKE Rev Feb '06 137
TITLE GAS ENGINE DESIGN
AUTHOR C.E. LUCKE TRANSLATOR
PUBLISHER D. VAN NOSTRAND CO. PRICE 3.00
LIBRARY - INSTITUTE - PUBLIC - MECHANICS -
DATE OF LAST EDITION 1905
TOPICS ESPECIALLY WELL TREATED 137
GAS ENGINES
FIG. 3
REVIEWED FEB. 1906
AUTHOR SUBJECT

FIG. 1. TAB CARD (Canary) FIG. 2. BOOKS—SUBJECT INDEX (Blue)
FIG. 3. BOOK REVIEWS—SUBJECT INDEX (Manila)

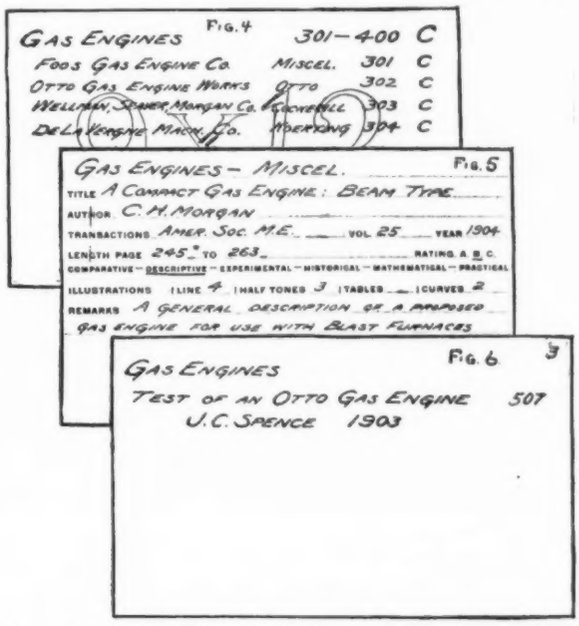

GAS ENGINES FIG. 4 301-400 C
FOOS GAS ENGINE CO. MISCEL. 301 C
OTTO GAS ENGINE WORKS OTTO 302 C
WELLMAN, SEAVER, MORGAN CO. CUYAHOGA 303 C
DE LA VERGNE MACH. CO. MORITZ 304 C
GAS ENGINES - MISCEL. FIG. 5
TITLE A COMPACT GAS ENGINE: BEAM TYPE
AUTHOR C.H. MORGAN
TRANSACTIONS AMER. SOC. M.E. VOL. 25 YEAR 1904
LENGTH PAGE 245 TO 263 RATING A B C
COMPARATIVE - DESCRIPTIVE - EXPERIMENTAL - HISTORICAL - MATHEMATICAL - PRACTICAL
ILLUSTRATIONS LINE 4 HALF TONES 3 TABLES CURVES 2
REMARKS A GENERAL DESCRIPTION OF A PROPOSED GAS ENGINE FOR USE WITH BLAST FURNACES
GAS ENGINES FIG. 6
TEST OF AN OTTO GAS ENGINE 507
U.C. SPENCE 1903

FIG. 4. CATALOGS (Salmon) FIG. 5. TRANSACTIONS (White)
FIG. 6. THESES—STUDENTS (Green)

To illustrate, turn to Gas Engines. Fig. 1 is a tab card. Back of it we find a blue card, Fig. 2, the numbers at the right of which refer to envelops, Fig. 3, which contain the written review of and other information which they may have about the books. The next is a salmon card, Fig. 4, with a list of gas-engine makers. The numbers refer to the catalogs. Fig. 5 is a white card with a reference to the Transactions of the American Society. Fig. 6 is a green card with a reference to a thesis by a student of the Institute. Then there are a number of amber cards referring to articles which have appeared upon the subject in different periodicals. Passing these we come

Much space is devoted in the present issue to the consideration of the papers and the ensuing discussions, but much remains yet to be said. The social features, aside from the usual reception at Sherry's, were a visit to the Mount Morris station of the New York, New Haven & Hartford Road, and a reception by the War Department with an inspection of the proving grounds at Sandy Hook and Fort Hancock.

The following officers were elected for the ensuing year: Prof. F. R. Hutton, president; John W. Lieb, Jr., P. W. Gates and Alex Dow, vice-presidents; Calvin W. Rice, secretary; W. H. Wiley, treasurer. The new members of the board of managers are A. J. Caldwell, G. M. Basford and A. L. Riker.

to the middle tab cards dividing the subject into smaller groups. Fig. 7 is the one for Design. Back of this we find an amber card, Fig. 8. Fig. 9 shows a blue card used for the author's index in connection with book reviews only. Fig. 10 shows a sample of the cards as found in the manufacturers' directory; Fig. 11 a sample of the cards from the graduates' index, giving the thesis number of each; Fig 12 is a sample of the labels used on catalogs.

The difficulty in inaugurating such a system would be the analysis and subdivision of the subject, and the paper is made complete by a presentation of their sub-divisions occupying forty-eight pages of the transactions.

WEIGHTS AND MEASURES.

Henry R. Towne argues in a moderate and well-considered paper and from the standpoint of an engineer and manufacturer against the efforts of the advocates of the metric system to force that system upon this country and advocates the creation by Congress of a Technical Commission on Weights and Measures to consist, in the discretion of the President, of not less than nine nor more than fifteen commissioners, to be appointed by the President by and with the advice and consent of the Senate, to investigate and report to the Congress upon the subject of measures of length, weight and volume in use throughout the United States and other countries, their investigations to include the question of retaining or modifying any or all of the present standards of length, weight and volume employed in the United States; second, in case present units are to be retained, the question of improving and simplifying, especially for the purposes of computation, the subdivisions and multiples of such units; and, third in case any or all of the present units are to be discarded, the desirability and expediency of substituting therefor the so-called metric units and the system based thereon. The act creating the Commission provides also that the President shall invite the Government of the United Kingdom of Great Britain and Ireland to participate in a conference for the improvement and unification of the standards of length, weight and volume used by the United States and the United Kingdom.

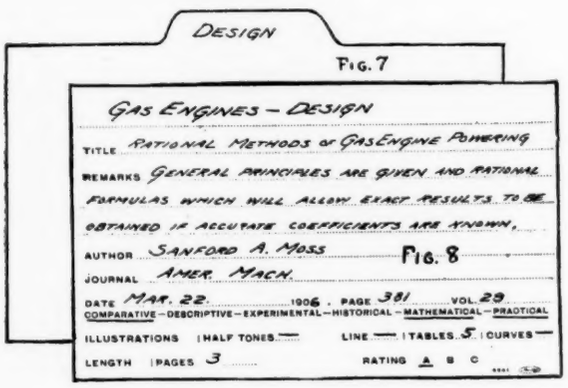

DESIGN
FIG. 7
GAS ENGINES – DESIGN
TITLE RATIONAL METHODS OF GAS ENGINE POWERING
REMARKS GENERAL PRINCIPLES ARE GIVEN AND RATIONAL FORMULAS WHICH WILL ALLOW EXACT RESULTS TO BE OBTAINED IF ACCURATE COEFFICIENTS ARE KNOWN.
AUTHOR SANFORD A. MOSS
FIG. 8
JOURNAL AMER. MACH.
DATE MAR. 22. 1906 . PAGE 381 VOL. 29
COMPARATIVE – DESCRIPTIVE – EXPERIMENTAL – HISTORICAL – MATHEMATICAL – PRACTICAL
ILLUSTRATIONS | HALF TONES — LINE — | TABLES 5 | CURVES —
LENGTH | PAGES 3 RATING A B C

FIG. 7. TAB CARD (Canary)
FIG. 8. PERIODICALS (Amber)

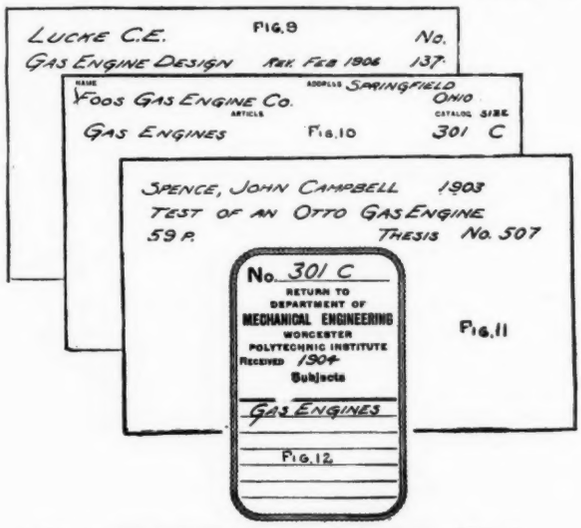


FIG. 9. BOOK REVIEWS—AUTHORS' INDEX (Blue)
FIG. 10. MANUFACTURERS' INDEX (White)
FIG. 11. THESES—GRADUATES (Cream-White)
FIG. 12. CATALOG LABELS (White)

THE FLOW OF FLUIDS IN A VENTURI TUBE.

Edgar Park Coleman presents the results of experiments showing that the Venturi tube is practicable for use as a meter to measure air or gas, and to some extent, steam. The paper explains the theory of the tube, considers its operation mathematically for both liquids and gases, and recounts experiments with steam, gas and air. With steam of more than a small percentage of entrained water the results were rendered uncertain by the varying value of the coefficient of discharge accompanying the increased wetness of the steam. With blast-furnace gas (and presumably with producer gas) a deposit adheres to the throat and other parts of this meter after continued use. The deposit, if not removed, affects the accuracy in two ways—first by decreasing the area of the throat, and second by closing or distorting the small holes leading to the pressure chambers.

The tests described, especially those with steam, do not cover a sufficient range of condition to be conclusive in every way. With better facilities, and to determine completely the limitations of the Venturi as a measuring device, further experiments should be made covering a more complete range of conditions for both gases and vapors.

IMPROVED TRANSMISSION DYNAMOMETER.

Professor W. F. Durand describes a dynamometer which may, in effect, be considered as a development of the six-equal-wheel form by the elimination of the lower two wheels and the substitution of chain for belt drive. The apparatus is shown diagrammatically in Fig. 13 and in perspective in Fig. 14, where A and B are two sprocket wheels mounted on a frame XY, which is carried at E by a steel spring supported in the manner of the well-known Emery steel plate knife-edge. This is frictionless, and for small movements of the frame the bending stress developed is infinitely small in comparison with the other forces in play. As shown at SS, semi-circular plates of steel are attached to the beam with slots at the bottom one-quarter of an inch in width to limit its range of motion. Sprockets C and D, similar in form and size to A and B, are mounted on standards attached to the base. All sprockets have ball bearings in order to reduce friction to the minimum. Around these wheels is led a chain, the dimensions being so adjusted that with the proper length of spring for the bearing at E, the chain throughout its length will run with the proper amount of slack for smooth and steady operation.

The shafts which carry the sprockets B and C are extended, carrying pulley wheels and also universal-joint couplings for direct connection to a driver or a follower. The power can be put in at C and taken out at E, or *vice versa*, and on either side of the dynamometer as desired. If the power passes through from C to D and C turns as indicated by the arrow, it is evident that the tight side of the chain will be from H around through I, J, K, L, M, and the loose side from M through NP to Q. Thus $H\ I$ and $K\ L$ will, under the higher or driving tension T_1, be equal on both sides except for the friction of the ball bearing on which B runs. Likewise $M\ N$ and $P\ Q$ will, under the same loose or following tension T_2, be the same on both sides except for the friction of the ball bearing on which A runs. Obviously the two tensions T_1 on the right, acting against the two tensions T_2 on the left, will de-

termine a moment tending to depress the end of a lever R attached to the frame XY, and the power is easily determined from this moment and the chain velocity.

THE EVOLUTION OF GAS POWER

was the title of an interesting paper by F. E. Junge, which he prefaced with the reminder that the early period of growth of a novel method of energy making [conversion?], as in the case of every other technical innovation, must be sporadic and encounter competition in order to thrive. The Diesel engine, he said, stands as an example of this fact. Conceived some twenty years ago, it was not made a commercial machine until quite recently, among the obstacles to its development being the difficulties in finding suitable materials that would stand the high temperature and pressure stresses. A similar experience was true of the Brayton engine, the working process of which Mr. Junge thinks is bound to find more general recognition in the future, if the ideal in gas engineering is to be realized, namely, "regulable and enforced combustion of a precompressed dynamic medium of air and fuel in quantities determined by the cut-off controlled by the engine governor according to the

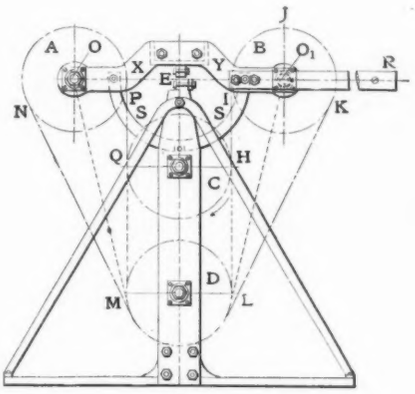


FIG. 13. DURAND'S DYNAMOMETER

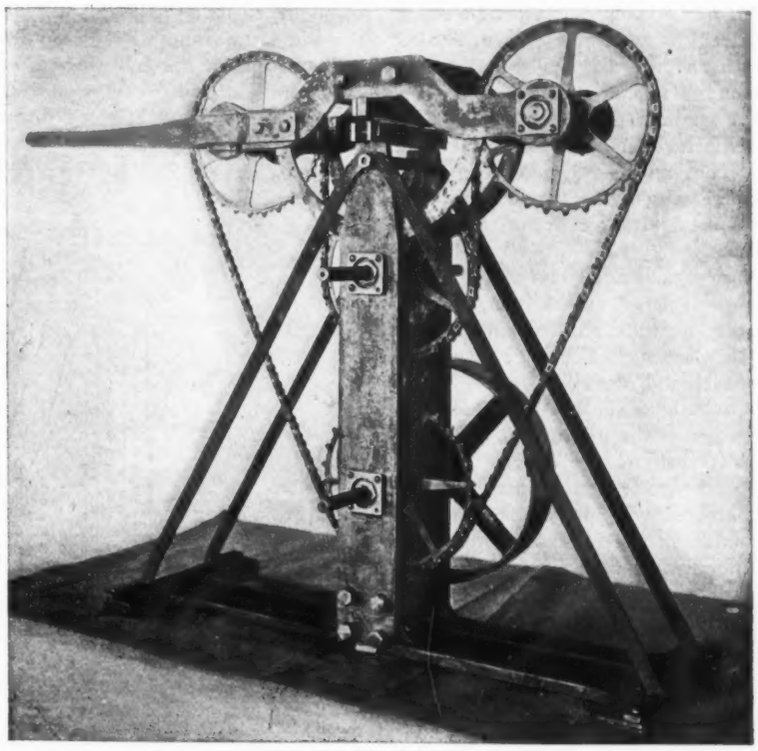


FIG. 14. DURAND'S TRANSMISSION DYNAMOMETER

load." But Brayton was hopelessly ahead of his time, and it is likely that even today there are but few competent observers who will admit that there is salvation beyond the present standard design.

For the ideal process of continuous combustion it matters not whether the fuel burned is gaseous or liquid. In order that combustion be brought about it is necessary that air and fuel should be introduced in correct proportions under similar cyclic conditions during the entire range of load. Further, the two constituents must be perfectly mixed when entering the cylinder, in order that each fuel molecule may find its corresponding quantity of oxygen which is necessary to support combustion. Finally, since combustion of the particles of the charge causes a rise of internal pressure, while the initial piston stroke tends toward its reduction, the rapidity of heat influx must bear a certain fixed relation to the piston speed in order that the two counteracting influences may be equalized and continuous combustion at constant pressure secured.

In the Diesel engine none of the foregoing conditions are brought about. We have a constant body of air to support combustion, a pressure of injected oil vapor which does not bear a fixed relation to the varying internal pressure, and, therefore, a speed of fuel influx which is irregular and in no way corresponds to the piston speed of that period. Nor does each fuel molecule on entering the cylinder find at once its corresponding quantity of oxygen. This feature retards ignition and flame propagation and makes the combustion a seemingly continuous one, though what actually occurs is afterburning. Yet to the casual observer, the Diesel engine appears to be of the continuous combustion type. Notwithstanding these deficiencies, the thermal results of the engine are so excellent that it is truly indicative of what we may expect on a further approach to the ideal process.

The Weidmann engine, designed with a view of utilizing the experience gained with the Diesel engine, is similar to that of Diesel in that gasified fuel is injected into a highly compressed body of air in the working cylinder, with the remarkable difference, however, that a corresponding amount of air is introduced with the fuel by a receiver piston moving at a rate corresponding to the speed of the working piston. The fuel and the air are so intimately mixed that combustion must occur regularly. Fig. 15 gives a theoretical diagram of the proposed process, from which the inventor calculates a thermal efficiency of 50 per cent.

Engines of the Otto type have been developed to a state of high perfection. Their deficiencies and incidental phenomena have been so far eliminated that the enthusiastic advocate of gas power is apt to overlook them entirely. Yet often indicator cards taken under identical conditions from the same engine, every one will reveal its fundamental weakness, namely, the impossibility of controlling the combustion, which is the most important function of the working process. The irregular and imperfect mixing of the charge constituents, the possibility of premature ignition and afterburning are drawbacks of the present working cycle of gas engines.

The various attempts to improve on the working process as carried out in standard engines (as by prolonged expansion, compounding, and water injection) have proved to be entries on the wrong side of the balance sheet. The drawbacks common to all of these so called improvements are increased bulk, weight, first cost, and negative work expended. The best method of increasing expansion is by high compression of the charge before combustion. The most economical way of reducing heat losses in the exhaust is by utilizing the same for raising steam in an exhaust boiler. As high as 160 pounds per square inch can be generated in this way. Generally 10 per cent. or more of the total output can be recovered from the exhaust of gas engines. Even the interesting experi-

ments of that distinguished authority, Dugald Clerk, in which he tried to improve on the working process by increasing the density of the charge before compression, have failed to effect any considerable advantage. The additional neutral gas he used, though it reduces temperatures all around, tends to retard the influx of heat, and thereby promotes afterburning and heat loss through the exhaust. The combustion process pure and simple, as used in the standard types of engines, gives the highest economic efficiency attainable with the Otto cycle.

The variety of earlier forms has now been reduced to two classes; (1) the double-acting tandem four-stroke-cycle engine, and (2) the double-acting two-stroke-cycle engine. The single-acting type of each is applicable only in smaller sizes. Each type has its disadvantages and each its special field of application. Vertical engines promise great savings in manufacture and of course in floor space. They must be developed in order to compete with steam turbines in space economy, and also for purposes of ship propulsion.

Of the various methods of regulation applicable in large gas engines, namely, the quality, quantity, and combination systems, the last named (as developed by Reichenbach and Mees) is superior to all because it reduces the heat value and the compression of the mixture at the lower loads less than do the others. Therefore, more regular and efficient combustion is secured with decreasing load and lean mixtures, especially when the point of ignition is automatically advanced.

Regarding the latest thermal performances of internal-combustion engines, attention is called to a 14-h.p. Marienfelde alcohol motor and a 70-h.p. Diesel oil engine, each showing on test an indicated thermal efficiency of 41.7 per cent.; a 20-h.p. Güldner gas engine running on city gas with 42.7 per cent. efficiency and a 500-h.p. Borsig-Oechelhäuser coke-oven gas engine with 38.6 per cent. efficiency. These figures refer to approximately full load conditions. The operating efficiency of the latest types of gas engines, based on the actual output of available work is, therefore, between the limits of 32 and 33 per cent.

The gas producer in its present form is a comparatively modern creation. Developed from the regenerative furnaces employed by Siemens and others in the process of steel making, it found early attention in England and France. The names of Dowson and Dr. Mond must be mentioned among those who took an active part in the vigorous development of earlier forms. The latter is especially known as having evolved a producer-gas system suited to the utilization of low-grade bituminous coal with the recovery of sulphate of ammonia as a by-product.

These earlier producers were of the pressure type. In them steam and air were blown through an incandescent fuel bed, and the gas thus generated was stored in a holder. Their sphere of application was considerably reduced by the invention of the suction producer, which was first developed by Benier in France, and by Körting in Germany.

Suction gas plants are very simple, safe, and reliable in action, and have found an extensive field of adoption in Europe and also a limited sphere of usefulness in this country. The lower grades of American fuels present characteristics less suitable for gasification than are possessed by the European coals. It cannot be said that the gasification of coking coal here is an accomplished fact. In Europe there is no lower limit in grade of fuels. At the same time another and no less important problem has been solved, namely, the cleaning of the gas from tar and other impurities which are formed during the transformation process. This is done by converting the tarry products into fixed gases in the producer proper, either by blowing or drawing the unstable volatiles through the principal zone or a second zone of combustion where they are transformed into stable constituents which do not separate from the gas when being cooled. No cleaning apparatus is required except an ordinary wet-coke scrubber and means for drying.

There is no standard for the general design and the constructive principles of gas producers, such as obtains with gas engines. This would be true even if fuel characteristics were identical all over the world, since there is as yet no accurate knowledge even of the most fundamental features, as, for instance, the form and dimensions of the producer chamber, kind of grate, material for firebrick lining, water cooling of producer walls, size and location of gas flues, manner of air admission, rate of gasification, depth of fire, stationary or revolving ash table, effect of automatic feed, internal or external boilers. In short, it may be said that almost everything in the design is done by traditional methods not based on adequate experimentation and data. Of course, there are three or four different systems solving all of these problems in a fairly satisfactory manner, but there is little scientific knowledge available as to which of these is the most efficient.

Some of the above problems have recently been solved to a certain extent for European fuels. Thus, it was found advantageous in using the lower grades of bituminous coal to make the producer proper of cast iron with water-cooled walls. This eliminates clinkering entirely. Further, it has been found that the continuously-revolving ash table is wasteful in fuel consumption, too much coal passing through unburnt, and that it disturbed the quiet action of the fire by constantly breaking up the numerous small gas passages in the fuel bed. Finally, it was found advantageous to have some preferably automatic control of the blast or of the suction pressure effecting gasification in order to be able to reduce or increase the rate according to the condition of the fire or the load on the engine or station.

In experimenting on this latter condition, it was found advantageous to insert between the gas engine and producer a fan the speed of which is automatically varied according to the engine or station load. This increases greatly the elasticity of the plant, making the producer capable of carrying heavy and continuing overloads without affecting the engine. At the same time it eliminates spare producer units, gas holders, pressure regulators, pulsometers, and other cumbersome and expensive apparatus, and keeps the supply of gas steady and of more uniform quality.

The gas holder, as an apparatus for improving on the quality and uniformity of the gas by promoting diffusion and separating out the water molecules, cannot be regarded as adequate, exposing, as it does, a large volume of gas to the varying and uncertain atmospheric influences of the season. As a storage tank for meeting fluctuations of load and peak

FIG. 2*

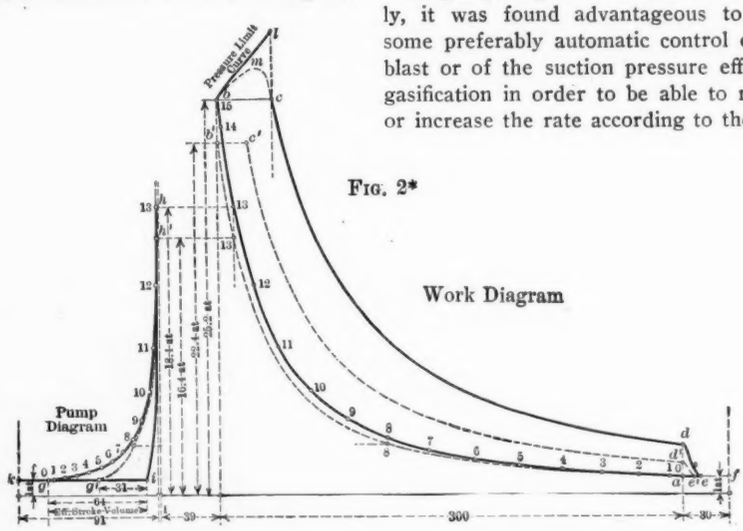


FIG. 15. THEORETICAL WORKING CYCLE OF WEIDMANN CONTINUOUS COMBUSTION ENGINE

loads of long duration, its value is problematical unless made very large, especially for the weak power gases. To keep a 500-h.p. engine running for 25 minutes the holder must have a capacity of 20,000 cubic feet. Long gas mains of ample section give sufficient reserve for equalizing the pressure during short periods of fluctuation.

For operating gas engines on board ship, producers must have means for keeping up the temperature in the producer while the engine is running at slow speeds or at rest, since otherwise it will not start up again on account of lack of suitable gas. This can be readily obtained by keeping up the rate [?] of gasification through the exhausting fan and returning the gas into the producer, where it is consumed again, there being practically no loss but that of the sensible heat of the gas radiating through the piping and, of course, the power required for driving the fan.

No producer can be regarded as up to date that does not embody means for automatically adjusting the amount of water or steam admitted with the air into the fire bed in fixed proportions according to the load, since without this arrangement the fire will grow dead at the lower loads and the engine will not be able to pull up to a higher load again when necessary.

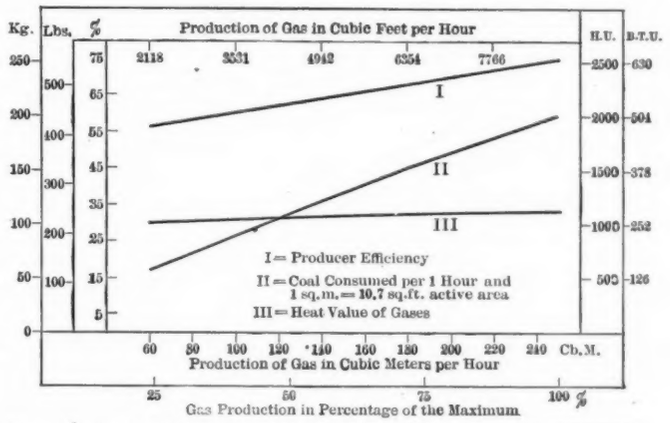


FIG. 16. PERFORMANCE OF DEUTZ DOUBLE ZONE LIGNITE PRODUCER

While the ordinary anthracite and coke producers show a general similarity, at least in type, the bituminous-coal producers and such as burn lignite and peat offer a striking variety of forms. Some take air in from below the grate, or from its circumference, or from a central pipe, and others have the fire on top and take air from above; still others have two fires and take air from both sides. With the double-zone producer as developed by Fichet and Heurty in France, and by the Deutz Motor Works and Körting Brothers in Germany, lignite and peat are the most desirable fuels to use, especially in the form of briquets. The raw fuels may also be burnt provided that they do not contain over 20 per cent. of water, since otherwise the upper zone of combustion is apt to gradually wander down and therefore a second grate has to be inserted. However, lignite and peat containing excessive moisture are of little importance beyond the field of their production on account of the high cost of transportation.

The double-combustion process in itself is very simple. In the upper layers the coal is transformed to coke, the unstable gases being discharged through the incandescent zone where they are transformed to fixed gases. The coke thus formed is burnt up in the lower fire after it has been extinguished for some time in the middle zone on its way downward. The gases formed below are, of course, stable and are drawn off together with those from above.

While the average consumption of anthracite and coke producers is about one pound per brake horse-power-hour and usually less, Fig. 16 shows the performance of a Deutz double-zone producer burning lignite as fuel. It is seen that the efficiency of the process drops from 75 per cent. at full load to 63 per cent. at half load. This is not a very good performance for a producer, generally speaking, since with superior fuels up to 85 per cent. efficiency can be attained. But it is of greater importance for the regularity of running that the composition of the gas remains practically constant at all loads, as is indicated by the curve. The economic results of these plants are excellent. An 80-horse-power plant using Bohemian lignite of 9200 B.t.u. per pound consumed on test 1.19 pounds per horse-power-hour, corresponding to a heat consumption of 10,950 B.t.u., and to a price of one-eighth of a cent per horse-power-hour in the particular location. (Meissen.)

The successful gasification of other low-grade combustibles, such as culm banks, is performed in the Jahns ring producer and others. The Jahns apparatus consists of a series of retort chambers containing the incandescent charge at different stages of the transformation process. Through these chambers the gas is drawn in succession. After the contents of one chamber have been completely gasified, it is separated from the rest, emptied, cleaned, and charged up again to enter the ring afterward as the youngest member. In the Von der Heydt coal mine at Saarbrücken, Germany, 2100 tons of culm have been gasified, giving out a total of 40,000,000 B.t.u., or 3245 B.t.u. per pound. The gas is used for driving gas engines and the plant has been in successful operation since 1903. The gasification of culm piles will develop, from every ton of culm charged in the producer in 24 hours, about 25 horse-power, after deducting losses through deterioration.

Of all the various industries which have been benefited by the evolution of gas power, the iron industry has been the most favored, since the blast furnace as the potential source of energy not only serves to convert the raw materials into pig iron, but also produces gas as a valuable by-product. This can be used for producing the power that is required in the power plant proper, and leaves available a considerable surplus for other uses. Of the total quantity of gas generated in a blast-furnace plant about 50 per cent. is required within the plant, while the rest, amounting to 25 horse-power per ton of pig iron produced every 24 hours, is available for outside purposes or sale.

In modern by-product ovens the quantity of gas produced depends on the quality of the coal coked, on its moisture and on the type of oven, and the composition varies considerably during one coking period. Deducting 60 per cent. for heating retorts and 10 per cent. for driving plant auxiliaries, there remain for every ton of coal coked in 24 hours from 5 to 6 horse-power available for other uses.

It has been found that the gain which can be effected from either the sensible heat of the gases in steam boilers, or from their sensible and their inherent heat, or from the combustion of all surplus gases in gas engines is approximately the same in the three types of coke ovens, namely, the beehive, the by-product, and the regenerative oven. In the latest regenerative ovens of the Otto type up to 4940 cubic feet of gas per ton of coal coked are obtained, the gas consisting chiefly of CH_4 and H, and having a calorific value of about 4000 calories per cubic meter (448 B.t.u. per cubic foot).

That the total amount of useful power that can be gained by scientific transformation from the first two sources alone is no negligible quantity, will be seen when applying the above figures to American conditions. With an annual coke production of 35,000,000 tons in the United States, and utilization of the coke-oven gases in regenerative ovens, there can be developed with modern gas engines in the neighborhood of 1,500,000

horse-power, if a best consumption of 8000 B.t.u. per brake horse-power is assumed. With an annual pig-iron production of 25,000,000 tons the surplus blast-furnace gases will generate in the neighborhood of 3,000,000 horse-power in gas engines.

This large amount of surplus energy can, of course, be utilized only when gas power is employed for driving all machinery within the works. In small countries like Germany, England, and Belgium the disposal of the available energy from iron-smelting plants and coal mines offers no difficulty, owing to the close concentration of industrial centers. The power is partly used for electric distribution to other works or mines which have no individual power plants of their own, and part is sent to neighboring cities for lighting and other purposes. In the majority of cases it is found advantageous to distribute the surplus energy in the form of electric current rather than as gas, though this practice cannot be generalized.

A third application which the surplus power has found in Germany, and which recommends itself for adoption especially in this country, is to drive electric railways for the transportation of raw materials and finished goods throughout the commercial distribution sphere of iron and steel works. Technically considered, this problem is very attractive, since the particular application affords a nearly constant outlet for the surplus power all the year round, because the production of iron and the consumption of energy balance each other.

Particular emphasis must be laid on the fact that the gases must be cooled, cleaned, and dried before being used in gas engines. With blast-furnace gas the dust and the moisture must be eliminated not only when the gas is used in gas engines but also, though to a less degree, when it is burnt under boilers or used for heating blast stoves, etc., since plant economy is thereby greatly increased. The cleaning apparatus giving the highest all around efficiency is of the centrifugal high-speed type, such as the ordinary fan and the Theissen washer. The power expended in cleaning the gas can be brought down to from 2.5 to 1 per cent. of the power obtained by the purified gas. The consumption of water varies from six to nine gallons per 1000 cubic feet. The distribution of town gas for individual power purposes, while not so much restricted to central location within the city cannot, without loss, be extended over wide territories. Moreover, at the present price of illuminating gas, it cannot compete in the field of power production with the independent suction-gas plant even if the latter uses such high-grade fuels as anthracite and coke.

Speaking more particularly of the generation of power from illuminating gas, Fig. 17 shows the increase of the number of horse-power delivered from city gas works, of the number of illuminating-gas engines operated, and of the medium capacity of these engines in percentage of the figures obtained each previous year, all data referring to the performances recorded in Germany. During the twenty years for which data are available, the number of illuminating-gas engines has increased more than sevenfold, the mean horse-power generated having grown in the proportion of 1 to 18.

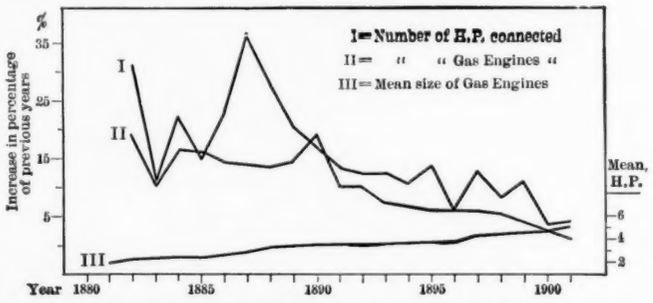


FIG. 17. SHOWING INCREASING APPLICATION OF GAS POWER FROM ILLUMINATING GAS IN 200 CITIES (GERMANY)

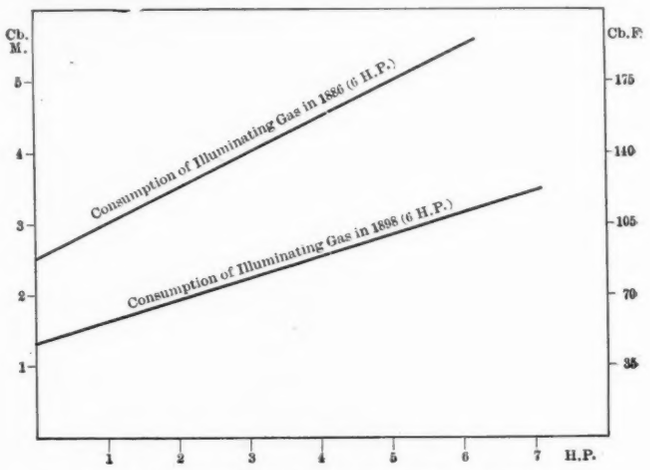


FIG. 18. CONSUMPTION OF KOERTING GAS ENGINE IN 1886 AND 1898

The economy of the illuminating-gas engine has in the meantime been increased as represented by the curves of Fig. 18, which show the comparative consumption of two 6-horse-power Körting gas engines, of which one was exhibited at the Karlsruhe exposition in 1886, the other at the exposition in Munich in 1898. Both charts together show how in the course of evolution the generation of power from city gas has grown more and more economical.

In the central part of Berlin, where real-estate prices and electricity-supply conditions are very similar to those which obtain in New York, there are quite a number of independent suction-gas plants which deliver electric energy to individual blocks. In Table I, the operating results obtained from January, 1904, to April, 1905, in one of these independent block stations are shown. It is seen that the average continuous fuel consumption per kilowatt-hour is about 1¾ pounds of anthracite, corresponding to a price of a little more than half a cent.

In metropolitan and other pumping stations suction-gas power has been adopted with great success, the duty per pound of lignite briquets being 1,270,000 foot-pounds in Germany. Table II shows the results attained with some of such plants in England.

TABLE 1. PERFORMANCE OF SUCTION-GAS PLANTS IN BERLIN, GERMANY.

Block.	Kw. Hours Rendered.	Coal Consumed, Lbs.	Coal Consumed per Kw. Hour, Lbs.	Total Coal Consumption Lbs.	Oil Consumed per Kw. Hour, Grains.	Coal Consumed per B.H.P. Hour, Lbs.
1	370,774	654,597	1.78	7,249	137	1.11
2	273,913	481,708	1.76	4,349	120	1.10
3	273,354	506,722	1.84	3,068	129	1.14
4	183,978	350,020	1.89	2,008	74	1.18

TABLE 2. PERFORMANCE OF SUCTION-GAS DRIVEN PUMPING PLANTS IN ENGLAND.

Location of Plant.	Pump Horse-power.	Head in Feet.	Lbs. of Coal per H. P.-Hour.
Welwyn Water Wks.	1.896	120	2.17
Paris-Plage.....	13.9	137.6	1.68
East Kent.....	13.64	255	1.3 coke
	12.96	330	1.28 coke

The portable suction-gas plant is coming into practical use on farms in Germany, enabling the farmer, instead of buying expensive gasolene or converting potatoes or other valuable matter by a costly process into fuel alcohol, to take sawmill refuse or the straw or hay or other vegetable matter growing on the farm, and feed it directly to the producer to be there gasified, generating the required power at no additional heat cost. In order to get equal work output with straw four times the weight of coal and forty times its volume must be burnt. In a 70-horse-power plant a horse-power-hour was obtained with only 2.31 pounds of straw and a little less of hay. Allowing in this particular case one cent per horse-power-hour for operating expenses

and interest, and a value of $4.40 per ton for straw, it was found that a horse-power-hour cost 1.26 cents with wheat straw and 1.14 cents with oat straw. With the best portable steam engines coal gives one horse-power-hour for 4 cents, and petrol gives about the same, while electric motors supplied from a distant hydraulic-electric plant cost in the neighborhood of 12 to 16 cents per horse-power-hour. Where the straw has no immediate value and is burned, as on some American farms in the West, the application of this form of gas-power generation is, of course, even more profitable.

Reference has already been made to the application of gas power for the propulsion of vessels. Ships equipped with suction producers and engines have actually effected a reduction of coal consumption to one-third of that of steam ships. This fact, together with the reduction in space occupied by engines and coal bunkers and the corresponding gain in cargo space, make the adoption of gas power one of the greatest importance for vessels which are to possess the maximum radius of action combined with the minimum cost of operation.

While there is little or no gain to be expected in bulk and weight of the engine alone, the gas producer occupies only about one-third of the space of a water-tube-boiler or one-eighth of that of a Scotch marine boiler, the dimensions depending on the grade of coal burned. In a 7000-ton gas-driven cargo steamer, the saving of cargo space effected was 13,000 cubic feet. The weight of a gas producer compared to that of a water-filled boiler of the type such as is installed in yachts and tug boats, is from one-fourth to one-fifth that of the latter. The amount of water needed for evaporation is about one-half pound per horse-power for a coal consumption of three-fourths pound. On a trial run a 70-horse-power gas tug consumed in ten hours 530 pounds of German anthracite against 1820 pounds of steam coal used by the competing 75-horse-power steam tug.

For the propulsion of larger vessels, the double-acting vertical two-stroke-cycle engine is the most promising type, since it gives steady and quiet motion with variable speed, quick starting under load, and almost instant reverse when compressed air is employed, such features being indispensable for successful ship operation.

The Deutz Motor Works of Cologne, Germany, who were the first to investigate the technical and commercial possibilities of gas ships, have fitted their suction-gas system on eleven vessels, the power of the various engines ranging from 35 to 90 horse-power. Recently they built two flat-bottom barges of 240 tons for river traffic, equipped with engines of 100 horse-power, of which one is doing active service between Cologne, Antwerp and Rotterdam.

During the year 5200 tons of load were carried, representing 1,950,000 ton-miles, which was done at a cost of about 25 cents per ton. Had the material been transported from Cologne to Rotterdam by the ordinary steamboats the tariff for transport would have been about 50 per cent. higher. Another barge of the same capacity is used for the haulage of goods on the Saarbrücken-Mühlhausen canal, making a round trip of 170 miles in 30 days, including 9 days detention and 9 days with light load. Under these disadvantageous conditions the cost of transportation by the suction-gas propelled craft is 33 per cent. lower than that of horse traction, while the boat makes eleven round trips during the year as compared with seven complete journeys which were possible by animal traction before the introduction of the new system.

In the discussion of Mr. Junge's paper George J. Rathbun took issue with Mr. Junge's statement that the gas engine's lack of overload capacity cannot be overcome because it is inherent in the cycle. He suggested as a remedy for the lack of overload capacity ordinarily exhibited by gas engines that an engine should be designed so as to give one brake horse-power hour on 8000 B.t.u. with a clearance so proportioned that the maximum indicated thermal efficiency is obtained at 26 per cent. below the maximum horse-power output; this will give the engine an overload capacity of 35 per cent., with practically uniform efficiency above full load. Mr. Rathbun thought, however, that efforts along this line would be barren of results unless the engine were equipped with automatically controlled ignition. This feature, he said, will perhaps have more influence upon the development of high-efficiency engines than any other apparent factor. Compression does not appear to be the most important factor in economy; there are engines on the market all having the same compression pressure and requiring from 8000 to 16,000 B.t.u. per brake horse-power-hour. Each particular engine has its limit of economical compression pressure; if compression is carried beyond that point, there is no increase in economy or in the mean effective pressure because the effect of the increased initial temperature is offset by the increased radiation to the jacket water. Mr. Rathbun presented a gas consumption curve plotted from tests of a 105-horse-power vertical three-cylinder engine of his design, which was equipped with variable ignition controlled by the governor. This chart showed a consumption of 8000 B.t.u. per horse-power hour at full load, and the curve was almost a straight line from this point up to 140-horse-power output, the heat consumption being slightly less than 8000 B.t.u. above full load.

J. E. Johnson, Jr., suggested that in view of the superior thermal efficiency and the lower maximum combustion temperature of the controlled-combustion type of engine, of which the Diesel is the chief representative, there were attractive possibilities in the transmission of combustible gas under such initial pressure that no additional compression in the engine cylinder would be necessary. The necessary precaution for making tight pipe-line joints to withstand the high pressure involved would not increase the cost of the pipe line in anything like the degree that the increased pressure would augment its transmission capacity. The efficiency of compressed-gas transmission as compared with electrical transmission, he stated, is very much higher than is generally supposed. Mr. Johnson took exception to Mr. Junge's statement that the rate of gas consumption by a gas engine is one-third of the rate required by a gas-fired steam plant, and asserted that modern water-tube boilers and cross compound condensing blowing engines with Corliss valve-gears will yield a horse-power with one-quarter of the quantity of gas required by the types of boilers and engines employed in previous years, and on which Mr. Junge's comparison was based. He thought the gas engine would compare with the modern gas-fired steam plant in the ratio of two to three, instead of one to three, in fuel consumption. Any attempt to consider the blast furnace as an adjunct to the power plant may be technically correct, but he thought it commercially foredoomed to failure, and cited as an example a large steel works where gas engines had been adopted for blowing purposes and driving electric generators. Three of the furnaces were connected to the gas main supplying this machinery, and two of these went out of commission temporarily, leaving the third one alone supplying the engines. This third furnace began to show symptoms of sudden ailments to which the blast furnaces are liable, and for the remedy of which a shut-down is necessary, but it was important to keep the plant running, and the superintendent did so until another furnace could be put into commission. By this time the badly-working furnace had reached such a condition as to require ten days of hard work and great expense to put it in order, and the loss of output during this period amounted to about $60,000, while the loss in fixed charges and profits was probably $20,000.

Prof. R. H. Fernald, who is in charge of the producer gas tests which have been in progress at St. Louis for the past two years under the auspices of the United States Geological Survey, presented some interesting results of preliminary tests made last year. These

results were plotted in the shape of charts and indicated that the rate of coal per square foot of fuel bed area decreased as the quality of the coal increased, the average rates during the tests having been 12½ pounds per hour per square foot of coal of 9000 B.t.u. heat value, 8 pounds of coal of 12,000 B.t.u. heat value, and 5½ pounds of coal of 14,650 B.t.u. The efficiency of the gas generator increased with the increased heat value of the fuel; in other words, the generator efficiency varied roughly in inverse ratio to the rate of firing. Professor Fernald pointed out, however, that the tests were not conclusive but merely preliminary.

Prof. H. Diederichs agreed with Mr. Junge's remarks concerning the Brayton cycle. The constant-pressure gas engine, he said, has never been developed to the extent that the promise of its cycle seemed to warrant. Although comparison of the formulas for thermal efficiency would show the efficiency of the Otto cycle to be higher than that of the Brayton constant-pressure cycle, this is not true in practice, because the operating limit of the Brayton cycle is not fixed by the compression pressure but by the maximum pressure occurring in the cycle. Professor Diederichs demonstrated by diagram that the conditions under which the constant-pressure cycle shows superior efficiency to the Otto cycle are considerably within the range of the actual conditions obtained in the Diesel engine.

Prof. R. C. Carpenter presented an interesting résumé of the work of George B. Brayton, inventor of the constant-pressure cycle. The Brayton engine in operation, he said, was comparatively noiseless and gave an indicator diagram similar to that of the Corliss steam engine. Brayton, however, fell into the error of designing his engine to operate

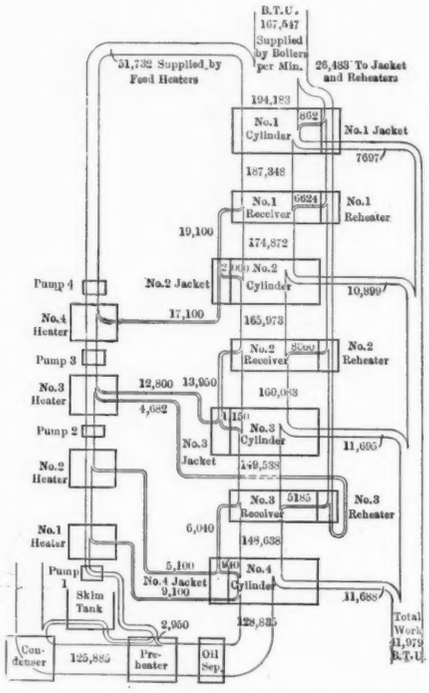


FIG. 19. FLOW OF HEAT THROUGH ENGINE

at relatively low maximum pressures, in consequence of which its economy was very poor. Professor Carpenter also pointed out that the Brayton cycle differs from the Diesel theoretical cycle in that ignition [combustion?] takes place at constant pressure, whereas it is supposed to occur at constant temperature in the Diesel engine. In practice, however, the Diesel engine more often operates on the Brayton cycle than on the theoretical Diesel cycle.

Prof. Arthur J. Wood said that although comparative figures in the appendix to Mr. Junge's paper [This appendix was an interesting but lengthy mathematical analysis not susceptible of abstracting.—Eds.] showed a higher efficiency for the Diesel than for the Otto cycle. This was due to the difference in the compression pressures of the comparison, Mr. Junge having taken the same maximum pressure for both engines. Had the Otto engine compressed to the same degree as the Diesel, this efficiency would have been higher. [Professor Wood apparently overlooked the fact that it is impossible to compress in the Otto type of engine to as high a pressure as that employed in the Diesel engine.—Editors.]

A HIGH-DUTY AIR COMPRESSOR.

Under the above title, O. P. Hood describes a compressor driven by a quadruple-expansion engine built by Nordberg, of Milwaukee, under a guarantee to develop 180,000,000 foot-pounds of work in the steam cylinder for each million of heat units used, a heat efficiency of over 23 per cent. The engine was to be supplied with steam of 300 pounds pressure, but the paper does not state any provi-

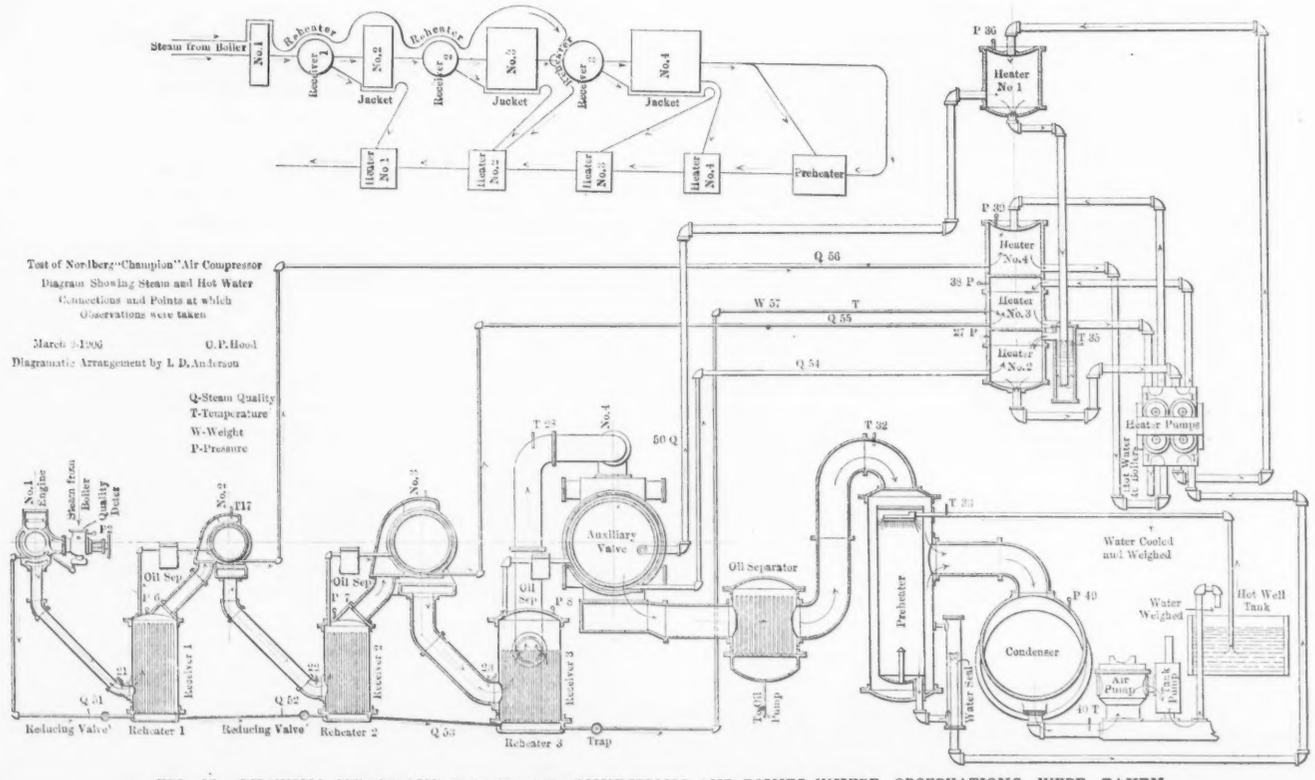


FIG. 20. SHOWING STEAM AND HOT-WATER CONNECTIONS AND POINTS WHERE OBSERVATIONS WERE TAKEN

sions as to its quality nor as to the amount of vacuum prescribed. As attached air pumps were used, the contractor probably undertook to get his own vacuum.

The boilers used proved inadequate to the 300 pounds pressure but sustained 250 practically without difficulty, and the test, reported in detail in the paper, was made at this pressure, and the necessarily reduced air pressure. Even under these conditions and with a vacuum of but 25.95 inches the engine made the astonishing record of 194,930,000 foot-pounds per million heat units supplied, beating the previous world's record, that of the Wildwood pumping engine by the same builder, by over nine per cent.

The efficiencies to different standards are as follows:

Heat consumed by engine per hour per i.h.p.	10,157 B.t.u.
Heat consumed by engine per hour per b.h.p.	11,382 B.t.u.
Equivalent standard coal consumption per hour (assuming calorific value of coal such that 10,000 B.t.u. are imparted to the boiler per pound of coal) per i.h.p.	1.016 lbs.
Per b.h.p.	1.138 "
Dry steam per hour per i.h.p.	11.23 "
Dry steam per hour per b.h.p.	12.58 "
Heat consumed per minute per i.h.p.	169.29 B.t.u.
Heat consumed per minute per b.h.p.	189.70 "
Efficiency of theoretical Carnot cycle between the temperature of incoming steam and the temperature corresponding to pressure in condenser	34.00%
Actual heat efficiency attained by this engine	25.05%
Relative efficiency with Carnot cycle.	73.69%
Relative efficiency with Rankine cycle	82.20%
Duty in work developed per million B.t.u.	194,930,000ft.lbs.

The engine owes its high efficiency to the principle, applied by its designer also in the Wildwood engine, of abstracting from the engine at different stages enough heat to raise the temperature of the feed successively to nearly that of the boiler. In the present instance, this process is divided into five steps, further heat being added by an economizer.

In order that the water might be weighed, it was taken from the fourth heater and cooled from a temperature of 334.5 degree to a temperature below the boiling point at atmospheric pressure. The progress of the temperatures is as follows:

	Source of Heat.	Temperatures. Water. Entering.	Water. Leaving.	Steam.
Preheater	Exhaust from low pressure cyl.	92.58	114.3	114.9
Heater No. 1	Low pres. cyl.	114.3	173.1	
" No. 2	No. 3 receiver	173.1	202.0	216.0
" No. 3	No. 2 receiver	202.0	269.7	291.2
" No. 4	No. 1 receiver	269.7	334.5	351.2

The accompanying engravings, Figs. 19 and 20, show the system diagrammatically, and the paper describes the engines and compressors and reports the test in detail.

BOILER AND SETTING FOR USE WITH BITUMINOUS COAL.

A. Bement, of Chicago, presented a modified design of boiler equipped with a chain-grate stoker, the objects sought being the attainment of a perfect and smokeless combustion and the full utilization of the boiler heating surface.

The first object is attained by roofing the furnace with refractory tiling suspended from the first row of tubes and by extending the furnace in front as shown in Fig. 21, so that the gases have time and space to thoroughly intermingle and completely burn before being chilled by contact with the heating surface. Fig. 22 shows one of the tile and Fig. 23 a series *in situ*. For the lower row 3½-inch tubes are used, affording greater space for the necks of the tiles than is available with the 4-inch tubes of the other rows.

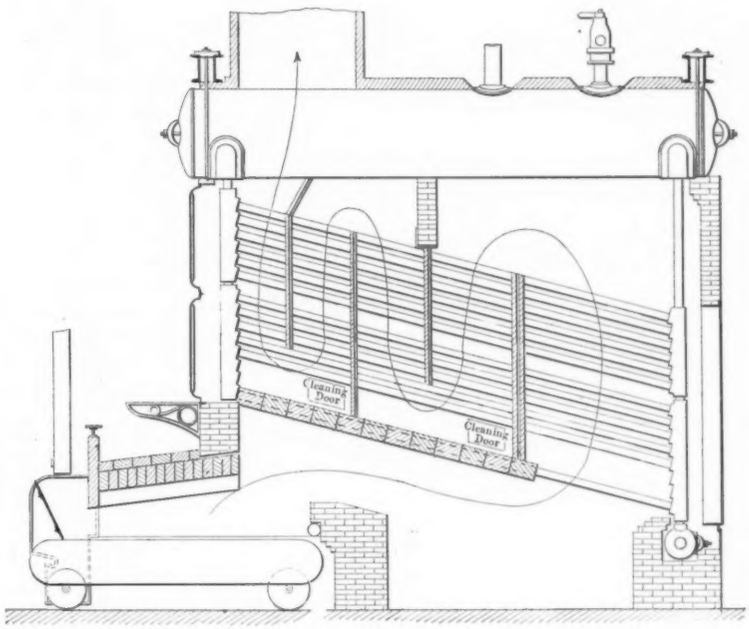


FIG. 21. BEMENT'S BOILER SETTING AND BAFFLES

To insure that all of the boiler surface shall be usefully employed, the gases are led repeatedly across the tubes by the vertical baffles shown, the passages decreasing in area as the gases contract by cooling. It is not practicable to make the cross-sections of these successive passes in the ratio of the volume of the gas without giving to the inlet a space which would reduce the length of the furnace roof. This compulsory flow brings the hot gases into contact with every portion of the heating surface and prevents the formation of pockets. It is in this respect that the setting now proposed differs from that described on page 744 of our December issue.

The contracted passages among the tubes produce a high resistance so that a strong draft is required; and the author recommends a combination of forced and induced draft (for the latter of which a chimney may serve) so arranged that the pressure in the furnace is that of the atmosphere, which prevents a leakage of air into the furnace through the front as when the furnace pressure is less than

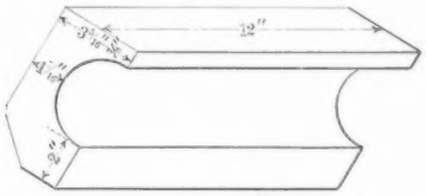


FIG. 22. A TILE FOR BEMENT FURNACE ROOF

that of the atmosphere, or the blowing out of flames and gas through doors and crevices when the internal pressure is greater. Photographs of chimneys showing the effect of the setting in preventing smoke are reproduced. Figures showing the effect upon the evaporation per square foot of heating surface will be interesting when available. The system is not patented and may be used by anybody who wishes.

REMARKABLE EFFICIENCY OF A DIMINUTIVE STEAM PLANT.

Prof. R. C. Carpenter describes in detail the 30-horse-power steam plant of a 1907 White motor car and reports a test made upon it under his direction at Cornell during the past summer, which

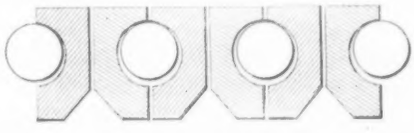


FIG. 23. SECTION OF FURNACE ROOF

presents results of interest to the steam engineer entirely aside from the special application for which the outfit was designed. The boiler consists virtually of a single coil of half-inch pipe through which the water is made to travel in a quantity automatically determined by the demand from the top to the bottom, thus reversing the ordinary order of things, and presenting a boiler with the water above the steam space. By the time the steam is drawn off it is highly superheated. The engine is a compound with 3- and 6-inch cylinders and a 4½-inch stroke, tested at 850 revolutions per minute. Gasolene was used as fuel. The average pressure at the boiler was 595 pounds; in the steam chest, 303 pounds. The average temperature in the steam chest was 757 degrees and of the exhaust, 239 degrees, showing that the steam remained superheated throughout the stroke. The average load shown by the dynamometer was 40.7 horse-power, the steam used per hour per brake horse-power, 11.96 pounds, and the heat per minute per brake horse-power 244 B.t.u. The actual evaporation per pound of gasolene was 10.15 pounds, and the average evaporation per square foot of heating surface per hour, 10.7 pounds. An independent test to determine the effect of a feed-water heater in the escaping gases showed an increase from 10.34 to 11.5 pounds of water evaporated per pound of gasolene.

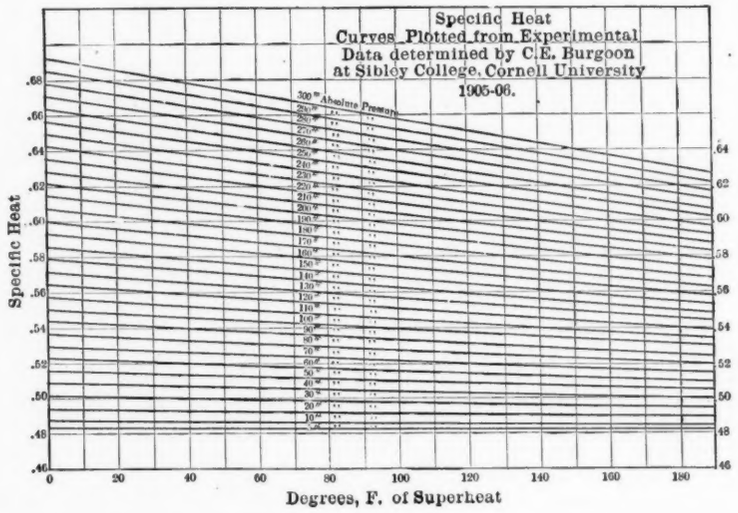


FIG. 24. SPECIFIC HEAT OF SUPERHEATED STEAM

SPECIFIC HEAT OF SUPERHEATED STEAM

Incidental to his paper on the small steam plant above described, Professor Carpenter alluded to the experiments which have been carried on at Sibley College at intervals during the last ten years and which are still in progress. Various methods have been tried and abandoned. Generally they were those which would measure the heat given off in passing from a higher to a lower degree of superheat, or *vice versa*, independently from the total heat in the steam, as in that way it was hoped to avoid the errors which are always caused by measuring the total heat which is great in comparison with that used in superheating. The first method employed was that of measuring the heat required for superheating by a known drop in pressure, being in many respects the reverse of the method used with the throttling calorimeter for measuring the quality of the steam. A second method tried was to measure the heat given off by the steam at constant pressure when being cooled from a high degree to a lower degree of superheat. The third method was the reverse of this, and consisted in measuring the heat required to raise the temperature from a lower to a higher degree of superheat by the electric current. The latter method has proven the most satisfactory, although great difficulty has been experienced in getting accurate and satisfactory apparatus for producing the necessary electric current uniformly and in obtaining suitable instruments for measuring the increase in temperature. The work has been largely done by Prof. Carl C. Thomas, C. E. Burgoon, and the testing department of the General Electric Company, of which Dr. W. R. Whitney is in charge. The temperature measurements were made at first with high-grade mercurial thermometers which, however, gave irregular results. Platinum resistance thermometers were then tried, which on test proved very satisfactory, but which later gave results which could not be checked. Nickel-steel constantin thermal-junction thermometers, which were made by Professor Shearer of the physics department of Cornell University, and calibrated by the use of their standards, were then substituted. These proved to be reliable and gave accurate results which could be checked by experiments made later. The experiments cover a range of pressures from zero to 220 pounds by gage and a range of superheat for these pressures to about 400 degrees Fahrenheit.

The results show an increase of the

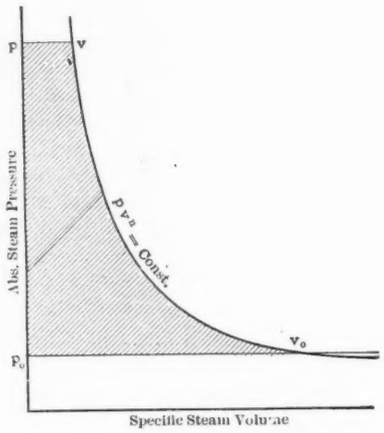


FIG. 26

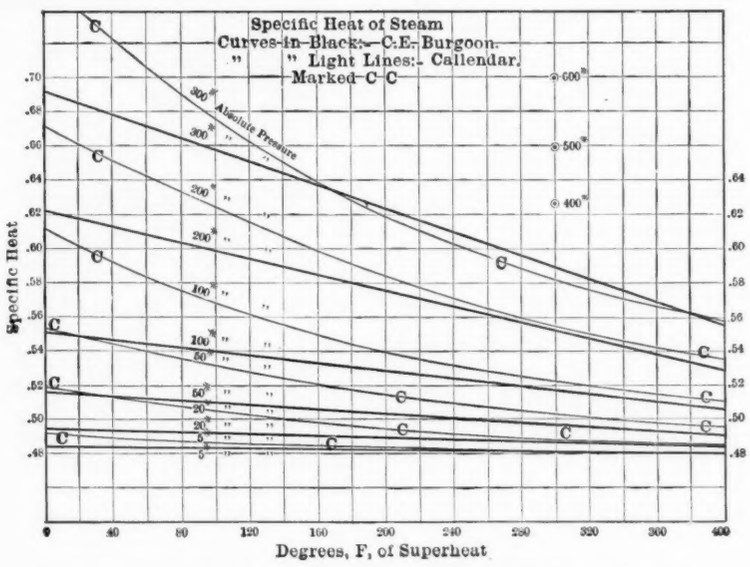


FIG. 25. COMPARISON OF SIBLEY COLLEGE RESULTS WITH THOSE OF PROFESSOR CALLENDER

specific heat of superheated steam with increase in pressure, and also a decrease in the specific heat of superheated steam with increase in the degree of superheat for constant pressure.

The experiments show that for a given degree of superheat, the specific heat increases directly with increase of pressure, as shown graphically in Fig. 24.

The experiments will be continued by Professor Thomas during the coming year through greater ranges of pressure and superheat, and the results will be presented in a paper which he will prepare for a future meeting of the society. The results which he has found are somewhat lower than those given by Professor Callender for low degrees of superheat, but essentially the same for high superheat. The Sibley results produced beyond the range of the experiment to high pressures and temperatures are shown in Fig. 25 compared with those of Callender. It has not been proven experimentally, however, that the specific heat continues to increase indefinitely with increase in pressure as shown in this diagram.

STEAM-TURBINE CHARACTERISTICS

Hans Holzwarth, the inventor of the Hamilton-Holzwarth turbine, presented a paper under the above title, describing a logarithmic diagram the application of which to turbine design is his own conception, and by means of which various characteristics of the steam turbine can be readily studied.

The usual pressure-volume diagram, Fig. 26, while it represents directly by the areas between the respective pressure lines the amount of available energy, and shows immediately the volume at any point of the expansion, has many grave disadvantages. The curve is difficult to construct, especially when, as for the adiabatic expansion of the turbine, the volume takes an exponent, and has to be laid out for each different initial pressure and volume. For superheated steam the curve is complicated by the change of the exponent at the point where the steam reaches the state of saturation. While equal areas represent equal amounts of work it is difficult to divide the diagram into equal parts or in a desired proportion, especially in the low-pressure portion where with the high ratios of expansion used the curve runs nearly parallel to the volume axis. It does not show at a glance what happens when the

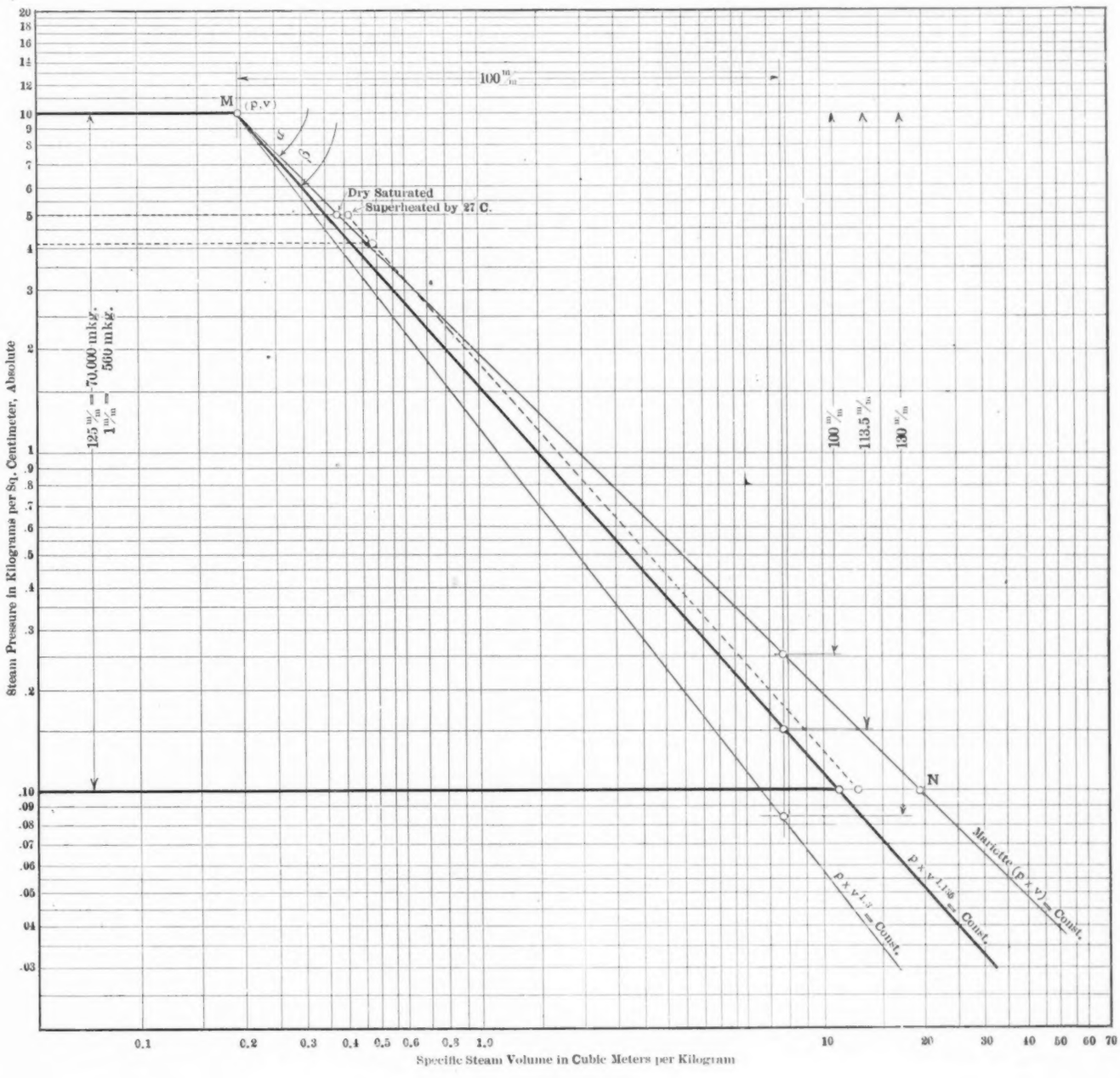


FIG. 27

initial pressure is raised or lowered; when the exhaust pressure is raised or lowered; when both these pressures are changed simultaneously; nor does it show at a glance the available total heat of the steam and the influence of moist or superheated steam.

If instead of plotting the diagrams with the pressures and volumes the logarithms of these quantities are used the expressions

$$pv = \text{constant and } pv^n = \text{constant}$$

become respectively

$$log.\ p + log.\ v = \text{constant},$$

and

$$log.\ p + (n\ log.\ v) = \text{constant},$$

the graphical expression of which is a straight line.*

Fig. 27 has been laid out in this way for one kilogram of steam expanding from an absolute pressure of 10 kilograms per square centimeter to one-tenth of a kilogram per square centimeter. The volume of a kilogram of steam of this pressure is 0.196 of a cubic meter and the logarithm of this number is set off to any convenient scale at a vertical hight proportional to the logarithm of the 10 kilograms pressures locating the point M. The volume at 0.1 of a kilogram on the pv=constant curve is easily found for $p_1 v_1 = p_2 v_2$ and

$$v^2 = \frac{p_1\ v_1}{p_2} = \frac{10 \times 0.196}{0.1} = 19.6,$$

the logarithm of which laid off upon the horizontal drawn at a hight proportional to the logarithm of 0.1 (the terminal pressure) locates the point N. A straight line joining these points is the equivalent upon this diagram of the pv = constant curve, and the straight-line equivalents of the exponential curves are very simply put in by multiplying the vertical distance of any point below the line of initial pressure on the curve by the exponent and making the distance on the same ordinate equal to the product. If α is the angle of the pv line and β the angle of a pv^n line, the tangent of $\beta = n \times tan.\ \alpha$.

The tangent of α in Fig. 27 is $cd = 100$, the tangent of β representing the curve for adiabatic expansion, the exponent of which is 1.135, is

$$1.135 \times 100 = 113.5$$

and by changing the tangent to this length the point through which the curve passes is located at e for the adiabatic and, by a similar process at f, for superheated steam where the exponent is 1.3.

The diagram may be laid out by using the logarithms of the pressure and volume (which can be obtained with sufficient precision from the logarithmic scale of the ordinary slide rule) upon ordinary uniformly ruled co-ordinate or cross-section paper; or by laying off the real pressures and volumes upon paper ruled logarithmically like that shown in Fig. 27.

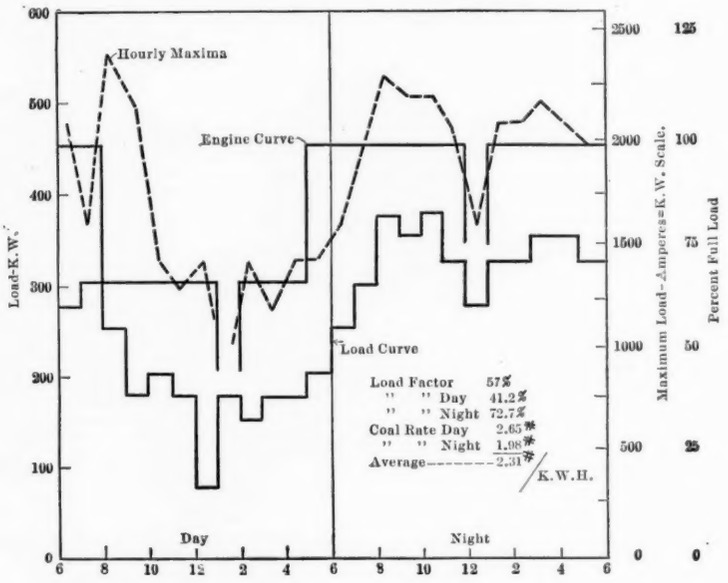


FIG. 28. LOAD DIAGRAM 450-KILOWATT GAS POWER PLANT

The amount of energy L in meter-kilograms developed by the expansion of one kilogram of steam between the pressures p and p_0 in kilograms per square centimeter absolute is

$$L = 10{,}000 \times p \times v \times (log.\ hyp.\ p - log.\ hyp.\ p_0),$$

The hyperbolic logarithm is found by multiplying the logarithm of the common table by 2.3025851. Mr. Holzwarth multiplies his constant by 2.31, reducing his formula to

$$L = 23{,}100 \times p \times v\ (log.\ p - log.\ p_0).$$

For example let

$p = 10\ Kg./cm^2 = 10{,}000\ Kg./m^2$.
$p_0 = 0.1\ Kg./cm^2$.
$v = 0.196\ meter$.

$$L = 23{,}100 \times \overset{p}{10} \times \overset{v}{0.196}\ [\overset{log.\ p}{1} - (\overset{log.\ p_0}{0} - 1)]$$
$$= 90{,}500 \text{ meter-kilograms.}$$

For one pound of steam between the limits of 100 pounds and 1 pound absolute (about 28 inches of vacuum) this would work out

$$1.44 \times 2.3025851 \times \overset{p}{100} \times \overset{v}{4.36} \times (\overset{log.\ p}{2} - \overset{log.\ p_0}{0})$$
$$= 28{,}913{,}100 \text{ ft. lbs.}$$

For adiabatic expansion $pv^{1.135}$ with dry saturated steam he gives the empirical formula

$$L = 270{,}000 \times \frac{log.\ p - log.\ p_0}{6.82 - 0.9\ log.\ p_0}.$$

As $6.82 - 0.9\ log\ p_0$ varies from 7.7 to 6.8 for values of p_0 from 0.1 to 1 kilogram per square centimeter this reduces to approximately

$$35{,}000 \times (log.\ p - log.\ p_0)$$

for the lower and

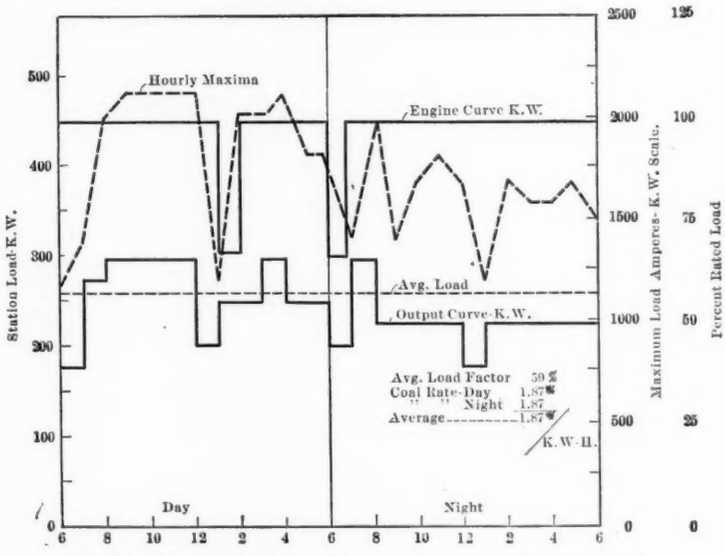


FIG. 29. LOAD DIAGRAM 450-KILOWATT GAS ENGINE PLANT

$$40{,}000 \times (log.\ p - log.\ p_0)$$

for the higher terminal pressure.

Within the above mentioned limits the amount of available energy L is directly proportional to

$$log.\ p - log.\ p_0,$$

that is, to the hight of the expansion trapezium. Where the expansions in the dif-

*See Straight Line Pressure-Volume Diagram, page

ferent stages only cover a range of three or four kilograms the factor

$$\frac{6.82 - 0.9\ log.\ p_3}{6.82 - 0.9\ log.\ p_2}$$

is so small that with limits accurate enough for practical purposes the areas of equal-expansion energy are represented in the logarithmical diagram by trapeziums of equal geometrical hight.

As soon as the value L of the total expansion is known the energy of each stage of the expansion can be directly measured with an ordinary scale. A slide rule is therefore the only tool required for operating with the logarithmic pv diagram.

The diagram reveals at a glance the following facts:

(A) By raising the exhaust pressure and correspondingly raising the initial pressure the total output of the turbine will be increased. Speaking in terms of the logarithmical diagram: by sliding the whole trapezium along the $pv =$ constant line upward, the available energy in one kilogram of steam remains the same, the available velocities in the individual stages remain the same, the specific volumes of the steam decrease, and consequently through the same vane area with the same velocities an increased weight of steam flows through the turbine, and, therefore, more energy is available at the switchboard.

(B) By lowering this trapezium along the pv^n = constant line, which means decreasing exhaust and initial pressures correspondingly so that the available steam energy remains constant, the total output at the switchboard will be decreased. The volumes in the individual stages being increased, and the steam velocities and vane areas remaining constant, a reduced weight of steam only can pass through the turbine, and, therefore, the output at the switchboard is decreased.

In both cases the total efficiency of the turbine is affected by the rotation and steam-passage losses, varying with the specific volumes of the steam. The latter operation will result in a better efficiency than the first one.

(C) Increasing the trapezium toward the exhaust-pressure lines, decreasing the exhaust pressure, and keeping the initial pressure constant means an increase of the available energy in direct proportion to the hights of the respective trapeziums, which results in an increase in steam velocities in the individual stages and increased specific volumes of the steam. Whether an increased weight of steam can flow through the turbine cannot be decided without reference to concrete figures; but whether the weight of the steam can be made greater or not, at any rate the output at the switchboard will be increased, the pressure upon the running vanes being heavier with increased velocity. The rotation and steam-passage losses will be decreased.

(D) Raising the trapezium at the initial-pressure line—increasing the initial pressure and keeping the exhaust pressure constant—results again in an increase of the available energy in direct proportion to the hight of the respective trapeziums. The velocities in the individual stages are increased, the specific steam volumes are decreased with the exception of that in the last stage. Therefore an increased weight of steam can travel through the turbine and the output at the switchboard is correspondingly increased. At the same time the rotation and steam-passage losses are greater.

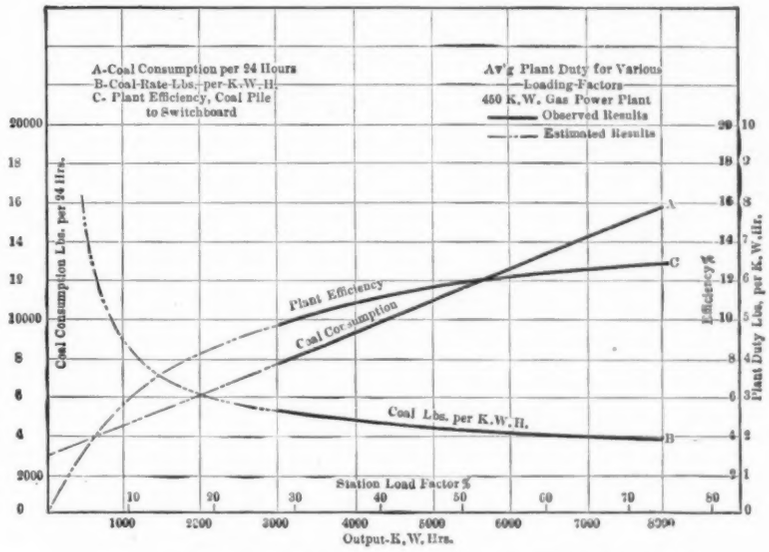


FIG. 31. AVERAGE PLANT DUTY AT VARIOUS LOADING FACTORS

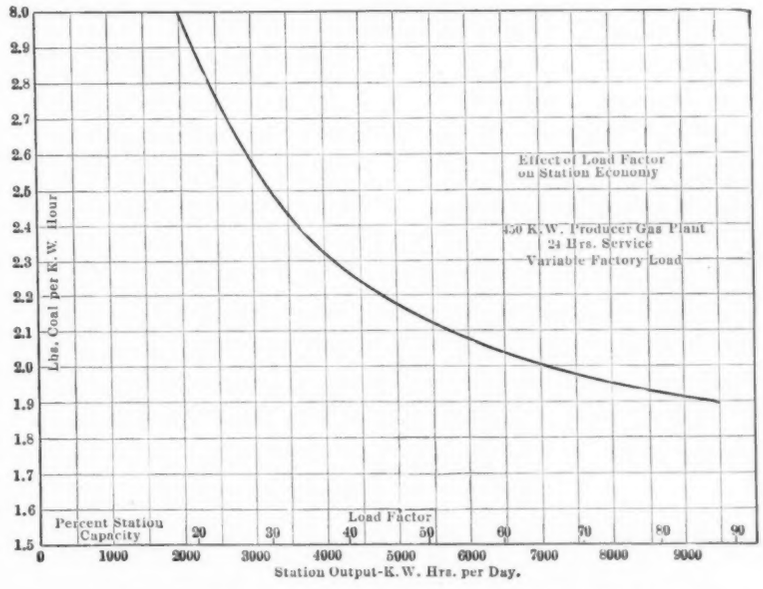

Effect of Load Factor on Station Economy
450 K.W. Producer Gas Plant
24 Hrs. Service
Variable Factory Load
Lbs. Coal per K.W. Hour
Percent Station Capacity
Load Factor
Station Output-K.W. Hrs. per Day.

FIG. 30. RELATION OF FUEL RATE TO STATION LOADING FACTOR

(E) Enlarging the trapezium in both directions—increasing initial pressure, decreasing exhaust pressure—corresponds to an increase in available energy in direct ratio to the hights of the respective trapeziums. The velocities are increased, the specific volumes partly increased, partly decreased; at any rate the same weight of steam will give a considerably greater output of energy at the switchboard.

(F) The question of the overload capacities of steam turbines can easily be studied with this diagram. The overload capacity of a steam turbine can be attained with or without by-passing steam to the low-pressure zone. Without by-pass steam it can be accomplished by raising the initial pressure and keeping the exhaust pressure constant (discussed under D), or by raising the exhaust and initial pressures simultaneously (discussed under A).

Admitting live steam to the low-pressure portion of the turbine does not necessarily involve a considerably larger output at the switchboard. This additional steam can be supplied to the original steam which has already worked in the

high-pressure zone, either with its full static pressure and no initial velocity (at least insignificant against the velocity of the original steam), or with a high initial velocity and a static pressure equal to that of the original steam at the inlet of the by-pass steam.

In the first case the by-pass steam exerts a back pressure upon the original steam equal to the static pressure of the by-pass steam. If this pressure is equal to the initial pressure of the original steam, this original steam cannot accomplish any useful energy in the high-pressure zone at all, and all the considerably increased rotation and steam-passage losses in this high-pressure zone have to be entirely compensated by the low-pressure zone.

In the low-pressure zone of course the amount of useful energy is considerably increased. The total available energy of the steam has to be converted into useful energy only in the low-pressure stage and therefore increased steam velocities allow an increased weight of steam to pass through the low-pressure zone.

When the by-pass steam is supplied with a static pressure equal to that of the original steam at the meeting point, but with a considerably increased steam velocity, then the high-pressure zone works just as before, initial and exhaust pressures being the same, and the low-pressure zone has the same available energy and in addition an increased weight of steam can pass through the low-pressure zone on account of the very great initial velocity of the by-pass steam. Therefore, the overload can be accomplished without sacrificing the high-pressure zone.

By tapping the individual stages at similar points and registering the static steam pressures, it is very easy by means of this logarithmical diagram to discover any irregularities in the expansion processes of these individual stages.

The vane and leakage areas can be easily checked with this diagram. For instance: two similar individual stages that are supposed to do equal work, have been checked as to their static initial and exhaust pressures; the values of these pressures are plotted in the logarithmical diagram on the *pv* line. The hights of the respective trapeziums show the ratio of the amounts of available energy in the individual stages. If these hights are not alike, the available energies differ in the stages. The reason for this must be that either the vane + leakage area is too small in the stage with the higher trapezium or the vane + leakage area too large in the stage with the lower trapezium.

While this logarithmical diagram answers in an almost perfect way all the requirements for a practical turbine characteristic, as far as the pressure conditions in the turbine are concerned, it cannot directly answer all the thermodynamical questions, for instance: questions as to the influence of moist and superheated steam, or as to the effect of throttling the steam with a regulating valve.

These questions can be answered with a very ingenious turbine diagram, first suggested by Professor Mollier, of Dresden, in *"Zeitschrift des Vereins Deutscher Ingenieure,"* 1904, p. 272.

The Mollier heat-entropy diagram is reproduced and explained and the author deduces from it that if steam of ten kilograms pressure is throttled down to five kilograms there will be a loss of available energy of only 13.7 per cent. and the steam will be superheated 27 degrees Centigrade. The effect of this is depicted upon the logarithmic diagram by locating the point on the 5-kilogram line but outside of the original *pv* line an amount sufficient to represent the increase of volume by the superheat. The superheated steam changes into saturated at 4.1 kilograms and from here on the line runs parallel with the original $pv^{1.135}$ line. This shows that by throttling the steam from ten to five kilograms less steam passes through the turbine on account of the increased steam volume and decreased steam velocities. The available energy is reduced 13.7 per cent. The rotation and steam-passage losses are decreased, therefore the total efficiency of the turbine will not be materially affected by this throttling operation, accounting for the maintenance of the efficiency over wide ranges of load.

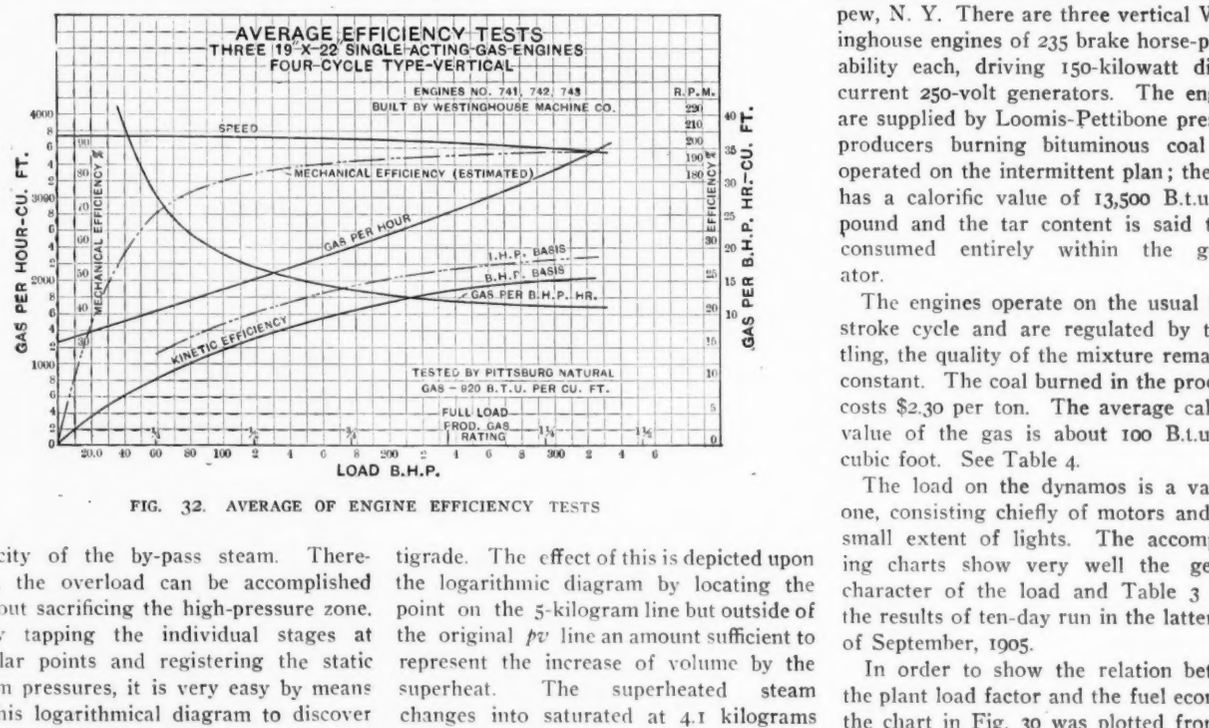


FIG. 32. AVERAGE OF ENGINE EFFICIENCY TESTS

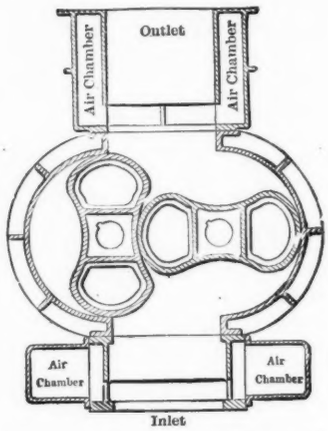


FIG. 33. CROSS-SECTION OF PUMP

An appendix, longer than the paper itself, gives specific examples of the application of the diagram.

OPERATING RESULTS FROM A MODERN GAS POWER PLANT

In a paper entitled "Producer Gas Power Plant" J. R. Bibbins gave some interesting figures taken from the regular operating log of a 450-kilowatt gas-engine plant in the Gould establishments at Depew, N. Y. There are three vertical Westinghouse engines of 235 brake horse-power ability each, driving 150-kilowatt direct-current 250-volt generators. The engines are supplied by Loomis-Pettibone pressure producers burning bituminous coal and operated on the intermittent plan; the coal has a calorific value of 13,500 B.t.u. per pound and the tar content is said to be consumed entirely within the generator.

The engines operate on the usual four-stroke cycle and are regulated by throttling, the quality of the mixture remaining constant. The coal burned in the producer costs $2.30 per ton. The average calorific value of the gas is about 100 B.t.u. per cubic foot. See Table 4.

The load on the dynamos is a varying one, consisting chiefly of motors and to a small extent of lights. The accompanying charts show very well the general character of the load and Table 3 gives the results of ten-day run in the latter part of September, 1905.

In order to show the relation between the plant load factor and the fuel economy, the chart in Fig. 30 was plotted from observations made at the plant under the various conditions stated. Fig. 31 was also plotted from observations taken during ordinary operation. The upper curve marked "Plant Efficiency" expresses the

ratio of work done to the heat in the coal, or the over-all efficiency, which is shown to be from 12 to 13 per cent. under average operating conditions, the corresponding load factor being 55 to 75 per cent.

Fig. 32 gives the average results of a number of efficiency tests made on the engines. From these and the data in Table 3 the author, by a process of elimination, gives the producer plant an efficiency of a little over 70 per cent. at half load.

Some trouble has been experienced with the gas gates in the piping, due, apparently, to accumulations of carbon on the seats. Straight producer gas is not made directly, but water gas and air gas are made alternately at intervals varying according to the richness of gas required, and the two are mixed in the holder. The proper cleaning of the gas also presents some difficulty because of the amount of lampblack in it; the lampblack does not adhere readily to wetted surfaces, as does ordinary cinder dust, and it is therefore more difficult to remove it.

The attendants were all unskilled before taking up this work, and the chief engineer expresses the opinion that, all things considered, he would regret to go back to steam.

TABLE 3. OPERATING DATA.

450-KW. PRODUCER GAS POWER PLANT.

Date, Sept., '05.	Engine Hours Run.	% Full Day Run.	Output Kw. H.	Load—Kw. Avg.	Load—Kw. Rated.	Station Loading Factor.*	Fuel Used.	Coal Per Kw. H.	Coal Per B.h.p. Hour.
20	71	98⅔	4850	202	450	45	11,100	2.29	—
21	65½	91	5275	220	450	49	11,400	2.16	—
22	70	97½	6550	273	450	61	12,700	1.93	—
23	45½	63½	4025	188	450	41.7	8,800	2.18	—
24 Sun	24	33⅓	2250	187.5	450	41.6	†6,600	†2.93	—
25	70½	98	6400	267	450	59	12,000	1.87	—
26	69⅔	97	6300	263	450	58	12,300	1.95	—
27	70½	98	6700	279	450	62	12,600	1.88	—
28	72	100	6700	279	450	62	12,900	1.92	—
29	70½	98	6600	275	450	61	12,900	1.95	—
Avg.	63	87½	5565	243.4	450	54	11,380	2.04	1.44

Westinghouse Vertical 3 Cylinder Engines—Loomis-Pettibone Producers.
* Loading Factor=% continuous generating capacity.
† Includes extra coke used on Sunday for starting new fires.

TABLE 4. CHARACTERISTICS OF POWER GAS.

TYPICAL ANALYSIS.

Date, June, 1905	20th	21st	23d
Number of hours test	3	10	10
Intervals of tests (minutes)	15	15	15
Minimum calorific value	98	91	91
Maximum calorific value	113	110	104
Average calorific value	105	100	100
Maximum variation from avg	7½%	10%	9%

Date.	H	CO	CH_4	O	CO_2	N
Sept. 9 1905	9.50	21.0	1.9	0.6	6.5	60.9
Sept. 12, 1905	10.13	22.8	2.2	0.4	5.6	58.9

TEST OF A ROTARY PUMP

Four years ago, John T. Wilkin presented to the society a paper on rotary pumps, in which he claimed that from many indicator cards taken from engines operating pumps of this character, he had found the combined efficiency of the pump and engine to be 80 to 84 per cent. The pump in question is the positive rotary of the two-lobed cycloidal type shown in Fig. 33.

These efficiencies, being the product of the efficiencies of the pump and engine, seemed to W. B. Gregory too good to be true, so he welcomed an opportunity to test such a plant located at Abbeville, La., and used to pump water for irrigating rice fields. The tests, which are reported in detail, show an average combined mechanical efficiency of the Hamilton-Corliss compound engines and the pumps, of 81.77 per cent. The paper also illustrates and describes the plant.

Casual Conversations

THE GAS ENGINE AND THE ELECTRIC MOTOR

"'Ave yer gorn ter sleep?" snorted the gas engine.

"Oh, no," hummed the motor pleasantly. "I've been working hard for the last forty hours."

"Well, I don't see yer movin' and I don't believe you're workin'. Just listen to *me*! Bang! Biff! Wallop! That's what I calls work."

"I can't reciprocate," smiled the motor, winking at the switchboard.

"Yer can't wot?" asked the gas engine, spitting on the tiled floor. He was a very common gas engine.

"Reciprocate."

"What's that?"

"Go backward and forward."

"In course yer can't," sniffed the gas engine, pityingly. "Look at me, how I ketch 'old of this 'ere crankshaft with my fist and twiddle it round—*that's* work, that is! Fetched all the blessed skin off my paw this mornin'; just gettin' a new skin on agen. Hope ter goodness *that* won't come off, eh?"

"Ah! now I reciprocate."

"Oh, yer do, do yer?" snorted the gas engine, looking down at him suspiciously. "Well, I can't see yer a-doin' of it, nohow!"

"I do everything invisibly."

"Then what's the good of yer? I likes a cove to be 'onest, I does. Wen yer a-doin' of a thing let 'em know as yer a doin' of it. Bang! Biff! Wallop! I ain't ashamed of what I does, I ain't."

Whereupon he spat another half-ounce of yellow mixture on the floor.

"What makes you jump in and out like that?" asked the motor, presently.

"Well, yer see, it's like this 'ere. Hevery other revverlooshun a charge of gas is busted behind my starn."

"Poor thing! And doesn't it make you hot?"

"Look 'ere, little un, who are yer a-pitying of? None of yer lip, now! As I was a-sayin a charge is busted behind my starn. Why, yes, it warms me up a bit—in fact, I might say I'm pretty considerable bloomin' 'ot at times. But, lor, it's the only way of doin' yer work well."

"I never got hot myself."

"No, I dessay yer don't, yer werry superior, clammy little reptile. But yer don't know what work is. Look 'ere now!—Oh, lor! Save us! Crikey!"

"What's the matter?"

"Crank gettin' 'ot agen! No, it ain't, its my fist! No, it's the crank! No, it's *both!* Oh, lor, here goes——"

Whereupon the gas engine began to squeak most pitifully. The driver came running up with an oil can, and at first he thought the engine had a pain in the stomach; but seeing a cloud of smoke over the crank he put his hand there to make sure. In his confusion he hurt himself so badly that the oil can went skidding across the floor, while the driver danced with pain and put his hand between his knees. Meanwhile, the gas engine shrieked as if his heart was being extracted, and finally pulled up with a prolonged groan of anguish.

"Some day," murmured the motor to himself, "they will organize a society for Prevention of Cruelty to Dumb Matter."—*The Electrical Bulletin*, London.

A tight belt is a power-consumer and a slipping belt is a speed-loser, but between the two there is a good point where you don't lose any speed and don't consume too much power.

ROTATION LOSSES IN STEAM TURBINES

With Chart Showing the Power Required to Overcome the Windage of Blades of Various Hights When Running at Various Velocities in Steam of Various Densities—The First Presentation of the Results of a Systematic Experimental Investigation of this Important Factor in Steam Turbine Engineering

BY HANS HOLZWARTH

The energy losses in steam turbines can be divided into two classes: (1) The losses that are independent of the load the turbine has to carry, i.e., the constant losses; (2) the losses that vary with the load, i.e., the variable losses.

In horizontal turbines only the friction losses in the journal bearings constitute the constant losses. These friction losses depend upon the dead weight of the rotary part and the circumferential speed of the shaft in the bearings, supposing that the rotary part is balanced well enough and the shaft is stiff or flexible enough to run without vibration. These constant losses are very small compared with the other losses.

The variable losses are: The friction losses in the thrust bearing; the rotation losses of the rotary part; the throttling, whirl, eddy and skin-friction losses of the steam on its way through the vanes; the leakage losses of the steam; the radiation and condensation losses of the steam.

All these variable losses depend upon the steam pressure in the different stages; and as this steam pressure changes with the load, the variable losses change with the load. The more load the turbine has to carry, the greater these variable losses will be as a rule.

This is the reason why the consumption curve of the turbines differs so advantageously from that of the reciprocating engine. The efficiency in steam turbines is nearly constant for a wide range of load conditions.

The rotation losses constitute an important part of the variable losses. The following article is offered as a contribution toward the solution of the questions: What influences the rotation losses in a steam turbine, and to what extent do the rotation losses depend upon these individual influences?

The rotation losses, or windage losses, or ventilation losses, as they are sometimes called, are equivalent to the amount of power that is required to drive the rotary part of a turbine through the steam of different pressures that surrounds the individual parts of the rotor.

In turbines of the drum type the rotor consists of two main parts: the smooth drum, with the smooth balancing piston, and the rows of vanes, that is, the interrupted part.

In turbines of the disk type the rotor consists of the smooth disks and the interrupted part, the rows of vanes.

The words "smooth" and "interrupted" are to be understood in such a way that they indicate whether the surface of the rotating body consists of an infinite number of consecutive circles, with centers in the axis of rotation, or whether the surface is interrupted in that respect.

It will be evident, without comment, that the interrupted parts of the rotor cause considerable more rotation loss than the smooth parts. For instance, a row of radially projecting vanes, with both edges free and ends uncovered, will cause more rotation loss than the same row of vanes with their ends covered with a band; and this row of vanes again will cause more rotation loss than a similar row of vanes, in which the ends and the edges on one side are smoothly covered; and this row of vanes again will cause more loss than a similar one, in which the ends and the edges on both sides are smoothly covered; that means then an uninterrupted smooth rotating body of the same outside shape as the original uncovered row of vanes.

Each single part of the rotor again will cause the more rotation losses, the greater its radial distance is from the center and the greater the number of revolutions per minute or the angular velocity.

The immediate stationary surrounding of the rotor influences the rotation losses too. The closer this stationary surrounding is with reference to the rotating part, the smaller the rotation losses will be.

And again the more irregular or interrupted the stationary surrounding is, the greater the losses will be.

The pressure or more exactly the density of the surrounding steam finally influences the rotation losses to a very great extent.

The individual influences on the rotation losses are, therefore, very various: (1) the geometrical character of the individual parts of the rotor, whether smooth or interrupted; (2) the geometrical character of the stationary surroundings of the individual parts of the rotor, whether smooth or interrupted; (3) the axial distance between the stationary surroundings and the individual parts of the rotor; (4) the radial distance of the individual parts of the rotor from the center; (5) the angular velocity or the revolutions per minute of the rotor; (6) the density of the steam surrounding the individual parts of the rotor.

To what extent do the rotation losses depend upon these six individual influences?

To solve this problem in an absolutely scientific way, it would be necessary to make the following experiments: Run smooth drums or disks of varying diameters, with vanes of varying radial hight, ends open and covered, stationary surrounding at varying distance, smooth and interrupted, with varying speeds in revolutions per minute in steam of varying densities, and find the net power required for overcoming the rotation losses alone.

All the numerous results, obtained from these experiments, would have to be condensated into a general mathematical rule, that could be applied then for all the different conditions.

From occasional experiments, carried on for some years with disks of different diameters, vanes of different radial hight, ends covered with a band, running at different revolutions per minute partly in air, partly in steam of different density, I obtained the following general results:

(1) Within limits accurate enough for practical purposes, the rotation losses are directly proportional to the steam density, or, to state it in a form a little less accurate, but handier for practical purposes, are directly proportional to the steam pressure.

(2) The rotation loss of smooth disks up to 50 inches in diameter, with vanes of radial length up to 2½ inches, ends covered with a band, running with speeds up to 4000 revolutions per minute in steam of different pressure, will be: Rotation loss in watts $= m \times$ absolute steam pressure in pounds per square inch, the variable m to be taken from the three-axial charts on the accompanying sheet.

To condense my various results into the least number of charts and yet give clear and exact readings, I found it necessary to use the three-axial system, as has been previously done in technical literature.

When using these charts it must be borne in mind that in fundamental experiments, all the ends of the vanes were covered with a band; furthermore, that the stationary surrounding was farther away from the rotary disk than will usually be the case in steam turbines. But as this axial distance between the stationary and the running parts varies even in the same turbine, and much more in turbines of different size, it will be wise to assume the most unfavorable conditions in this respect.

The values of rotation losses computed with my charts will be found in some cases of very small clearances to be too high by about 50 per cent., but as a rule they will give very fair practical results.

Chart I gives the values of m for disks of diameters varying from 0 to 50 inches, for speeds varying from 0 to 4000 revolutions per minute and for vanes ½ inch high.

Chart II, same, but for vanes 1 inch high.

Chart III, same, but for vanes 1½ inch high.

Chart IV, same, but for vanes 2 inches high.

Chart V, same, but for vanes 2½ inches high.

Although the method of using these charts is very plain, the computation of an example may aid in understanding them.

EXAMPLE: Disk diameter 30 inches, vanes 1 inch high, speed 3000 revolutions per minute, steam pressure 50 pounds per square inch absolute. Chart II will give the rotation losses. Follow the "Disk Diameter" axis to 30 inches, then the parallel to the revolutions per minute axis to 3000, then the parallel to the m axis until meeting the full line curve starting at 30. This vertical distance corresponds with the variable m. The scale for the variable m is given on the vertical main axis.

In our case $m = 162$. Therefore the rotation losses will be equal to $162 \times 50 = 8100$ watts.

To find the values of m for disk diameters not specially marked in the charts, I have given the dotted curves starting at the revolutions per minute axis.

These curves give for a *certain* speed the variable m in function of the disk diameters, while the full line curves give for a certain disk diameter the variable m in function of the revolutions per minute.

To find the rotation losses for radial hights of vanes between those given in the charts, it will be sufficiently accurate to interpolate between the next lower and higher values.

Chart VI gives the variable m as a function of the disk diameter and the hight of the vanes for 1800 revolutions per minute.

This chart demonstrates very plainly the importance of distinguishing between smooth and interrupted parts of a rotor.

Besides this it demonstrates that the total rotation losses in steam turbines have to be attributed largely to the rows of vanes; the smaller balance is due to the rotation losses of the smooth disks or the drum.

With the aid of these charts it will be a comparatively simple matter to come to concrete figures in comparing the different designs of steam turbines with special reference to their rotation losses.

In opening the water glass connections after putting in a new glass, it is generally recommended that the pet cock at the bottom of the glass be opened first and then the steam valve at the top, thus allowing steam to blow through the glass and warm it gently. Next open the water valve and close the drain at the bottom of the glass. If water is admitted first, and then steam, the glass will often break, because the water is much cooler than the steam, and there would be a greater expansion in the upper half of the glass than in the lower half.

DESIGN, CONSTRUCTION AND APPLICATION OF LARGE GAS ENGINES IN EUROPE—XV

Test of a Koerting Engine—Independent Producers Compared with Mond Gas and Steam—Suggestions for the Improvement of the Two-cycle Engine—The Significance of Comparative Tests

BY FRANZ ERICH JUNGE

In Figs. 107, 108 and 109 the results of a series of tests made on a 600-horse-power Körting engine of the type discussed in the preceding articles are presented. The engine was running on producer gas made from anthracite coal. The dimensions of the principal engine parts are given in Fig. 107, which also shows the various other items of interest, namely, net indicated horse-power, N; effective horse-power, N_e; total pump work, $N_i + N_p$; indicated horse-power of gas and air pumps respectively, efficiencies, etc. While the curves show the actual values of these items under varying load conditions, the data designated under tests *A, B, C* and *D* give their mean value, and the results marked at the bottom give the average of tests *C* and *D*. The curves bring out very clearly what relations the various items, such as the total pump work, bear to the indicated and the effective horse-power output. Of course, it must be remembered that the engine tested was of the original Körting type, not embodying any of the various improvements in construction which have been introduced since, and which have effected considerable savings, especially in the loss by negative pump work.

Fig. 108 shows the consumption of coal at various loads between the normal and the maximum. The mean indicated work averaged 779.4 horse-power, or with an assumed mechanical efficiency of 78 per cent., 608 effective horse-power. The average consumption of coal at this load was 0.787 pound, or a little over ¾ of a pound per effective horse-power. In studying these data sight must not be lost of the fact that they refer to metric horse-power, and that one metric horse-power = 0.986 English horse-power. With a 13,000 B.t.u. anthracite coal at $5 per ton the consumption in the producer was about 10,000 B.t.u. and the cost 0.22 cent per effective horse-power-hour. With anthracite at $3.50 per ton the cost per brake horse-power-hour was 0.13 cent. In all cases it is less than one-quarter of a cent under the assumed load conditions.

Since in modern European practice lignite and peat are the fuels used for the generation of power in gas engines—on account of their cheapness, general availability and other desirable characteristics—the cost per brake horse-power-hour is even lower. Comparing the present figures, attained with an independent suction-gas plant and anthracite as fuel, with those obtainable with gas engines supplied from large Mond gas plants burning low-grade bituminous coal, it is found that the results are very encouraging for independent operation. If Mond gas is sold at 4 cents per 1000 cubic feet the best figure for the cost of fuel per brake horse-power-hour is 0.42 cent, while if it sells at 8 cents, the cost is 0.84 cent. A high-class modern steam engine of the same capacity and working under similar load conditions, in order to be able to compete with the engine under discussion, would have to burn under the boilers slack coal costing $1.80 per ton delivered at the boiler house, and even at that the cost per horse-power-hour would be 0.26 cent at best. If the comparison were made on the basis of the respective plant fuel costs, then the stand-by losses caused by intermittent working and varying plant load factor would put the steam plant even at a greater disadvantage. I shall consider this question in a separate paragraph. In the particular case here presented it is clear that even in localities where lignite and peat are not available, or where producers for the successful gasification of these fuels have not yet been developed, and where anthracite or coke must be used as producer fuel, a reduction in fuel consumption of 50 per cent. or more can be secured through the adoption of an independent gas-power plant, against what is attained with any other form of power generation.

Fig. 109 shows the diagrams taken from power cylinder, gas pump and air pump, indicated at both ends of the cylinders. The mean indicated pressure of the front end of the power cylinder is 60.6 pounds, that of the back end 64.2 pounds. The corresponding pressures in the gas pump are 2.9 and 2.6 pounds, and in the air pump 4.8 and 5.1 pounds respectively On the diagrams showing the internal working process of both ends of the power cylinder about ten cards have been taken under identical load conditions. It is evident that while the superimposed compression and expansion curves match each other quite regularly, the combustion lines, which represent the period of heat influx and development, show considerable deviation. This irregularity of combustion becomes the more remarkable the

larger the engine and the larger the space which the flame has to traverse from the point of ignition throughout the mass of gas. It is quite certain that the future development of gas engines will do away with this weak point in the working process of the internal-combustion engine, together with the other unfortunate characteristics, namely, premature ignition afterburning and back-firing, all of which are drawbacks inherent in the cycle. By proper construction they can be reduced to a minimum and their harmful effects on the operation of the engine be reduced, but they cannot be eliminated entirely. This can only be done by taking recourse to a new continuous-combustion cycle, or rather by reviving the old Brayton cycle, and providing novel means for enforcing combustion.

The relative thermal merits of the different cycles (Otto, Diesel, Weidmann) and the prospects for future development were discussed in a paper on "The Evolution of Gas Power,"* published in the November *Proceedings* of the American Society of Mechanical Engineers, to which reference may be had for further information on these questions.

Study of the test reports here given together with what was said in the preceding paragraphs, will serve to clear up very effectively the much discussed question of two-stroke-cycle efficiency and pump work. In this series of articles I have attempted to discuss impartially the merits and demerits of both types of gas engines, not only as regards their pres-

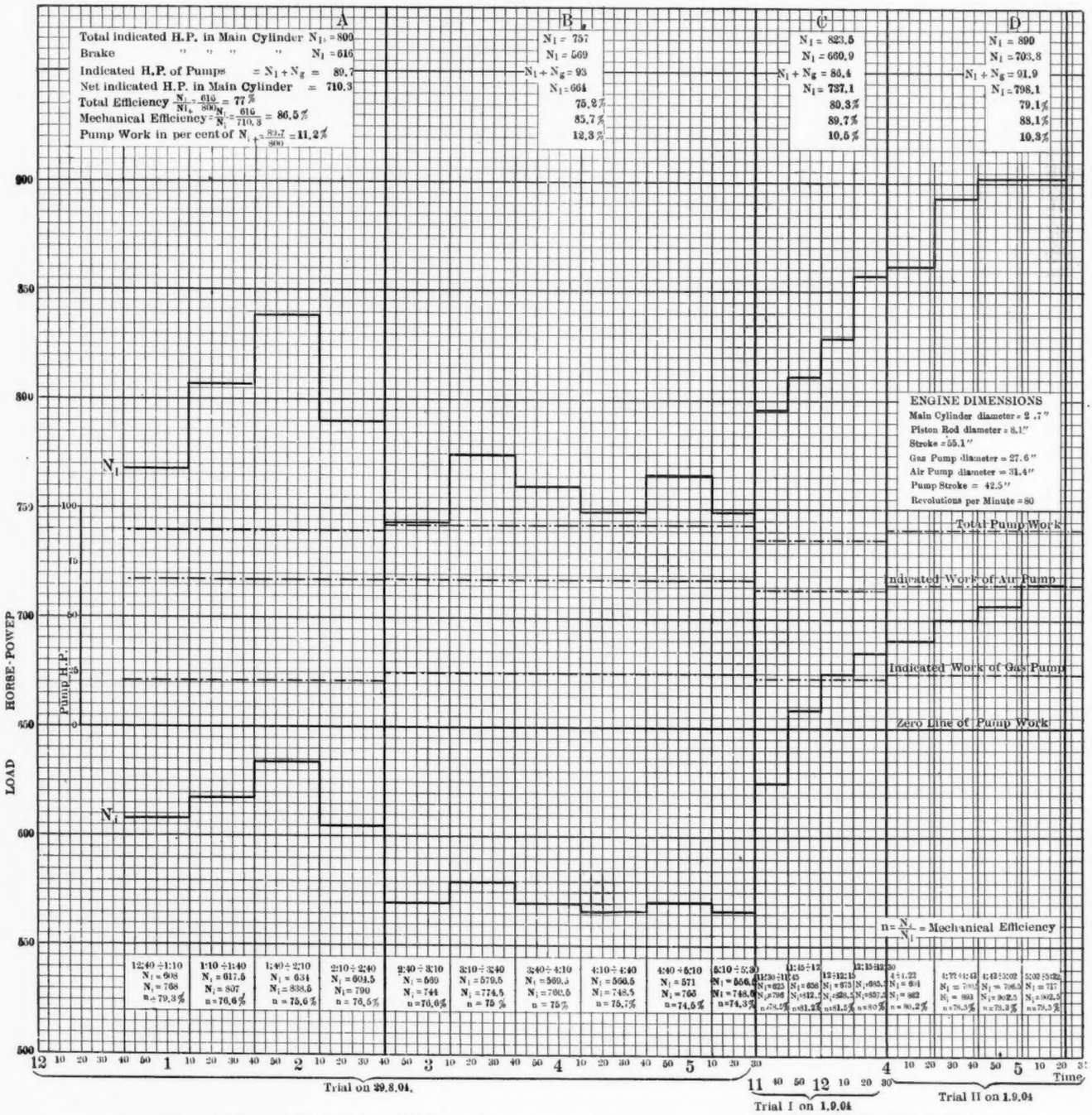

A
Total indicated H.P. in Main Cylinder N_{I_1} = 809
Brake " " " " N_1 = 616
Indicated H.P. of Pumps = $N_1 + N_g$ = 89.7
Net indicated H.P. in Main Cylinder = 710.3
Total Efficiency $\frac{N_1}{N_{I_1}} = \frac{616}{809} = 77\%$
Mechanical Efficiency $\frac{N_1}{N_i} = \frac{616}{710.3} = 86.5\%$
Pump Work in per cent of N_{I_1} $\frac{89.7}{809} = 11.2\%$
B
N_{I_1} = 757
N_1 = 569
$N_1 + N_g$ = 93
N_1 = 664
75.2%
85.7%
12.3%
C
N_{I_1} = 823.5
N_1 = 660.9
$N_1 + N_g$ = 86.4
N_1 = 737.1
80.3%
89.7%
10.5%
D
N_{I_1} = 890
N_1 = 703.8
$N_1 + N_g$ = 91.9
N_1 = 798.1
79.1%
88.1%
10.3%
ENGINE DIMENSIONS
Main Cylinder diameter = 2 .7"
Piston Rod diameter = 8.1"
Stroke = 55.1"
Gas Pump diameter = 27.6"
Air Pump diameter = 31.4"
Pump Stroke = 42.5"
Revolutions per Minute = 80
Total Pump Work
Indicated Work of Air Pump
Indicated Work of Gas Pump
Zero Line of Pump Work
$n = \frac{N_1}{N_i}$ = Mechanical Efficiency
HORSE-POWER
LOAD
Pump H.P.
Time
Trial on 29.8.04.
Trial I on 1.9.04
Trial II on 1.9.04
Mean Value of trials I & II of 1.9.04
Total indicated H.P. in Main Cylinder = N_{I_1} = 853.8
Brake H.P. " " " N_1 = 682.4
Indicated H.P. of Pumps $N_1 + N_g$ = 89.2
Net indicated H.P. in Main Cylinder = 767.6
Total Efficiency = $\frac{N_1}{N_{I_1}} = \frac{682.4}{853.8} = 79.7\%$
Mechanical Efficiency = $\frac{N_1}{N_i} = \frac{682.4}{767.6} = 89\%$
Pump Work in per cent of N_{I_1} $\frac{89.2}{853.8} = 10.4\%$

FIG. 107. RESULTS OF TESTS ON A 600-H.P. KOERTING ENGINE

*Abstracted elsewhere in this issue.—Eds.

ent construction and capability of competition with each other, but also as to future improvements which may be introduced. In addition to the improvements already referred to I mention the following suggestions, which were made relating to the construction of the Körting engine:

Cone-shaped piston head, forming a curved path for the outrushing exhaust gases and for the entering scavenging air, in order that the flow of gases and the change from axial to radial direction may proceed smoothly and without producing eddy currents or interruptions in the even flow. A cylinder composed of three parts, one middle portion containing the water-cooled exhaust slots and forming a cylindrical chamber to which are flanged the two symmetrical cylinder liners. Means for regulating the quantity of gas *and* air delivered according to the load, in order to decrease the negative pump work at low loads. A gas pump having a much larger clearance or compression space than the air pump in order that the influx of gas may be proportionately postponed.

BEARING OF COMPARATIVE TESTS

In the preceding articles of this series repeated reference has been made to the comparative cost of power, so far as the expenditures for fuel are concerned, from various sources, for instance, to the relation between independent suction-gas plants, Mond gas plants, and steam plants. The figures advanced in this connection are merely intended to give some rough idea of what enormous savings in fuel consumption can be made by the adoption of gas power, without taking into consideration the numerous other advantages, such as the prevention of smoke, etc., and without pointing to one particular engine system or plant. References of this kind are quite valuable in emphasizing the superiority of a modern power outfit over what can be attained by traditional methods of generation. To designate these comparisons as documents of the industry, or to attach to them exaggerated importance or scientific value is a mistake, because not only do they relate to conditions of actual practice only in rare cases, but they are mostly inaccurate and of momentary worth, owing to the constant shifting of the underlying economic agencies.

Referring first to the acute question of gas versus steam, it is almost impossible to find, without construing and adjusting, two analogous cases in which all the conditions affecting the commercial economy coefficient of a heat power plant are absolutely identical. Rarely can one find a gas and a steam-engine installation bearing the stamp of similar age and degree of perfection, a state which should preferably obtain in order to arrive at just conclusions. We are liable to quote figures obtained with the latest design of engines without justly discriminating between the types of steam engines available for comparison, whether condensing or non-condensing, whether working with superheat or without, whether employing economizers or other accessories of recent date. And even if we diligently mark down all these distinctions, then the valuation of the superiority claimed for one system over the other will yet be strictly a matter of individual appraising.

Even if we confine ourselves to the determination of the one item of respective plant fuel cost there are unavoidable differences in every instance which render the results of such investigations rather problematical. Assuming the maximum capacities of the respective gas and steam prime movers to be identical, and that they carry an ideal load which remains constant during the 24 hours of a daily run, even this apparent similarity of conditions does not suffice as a basis for comparison, since the process of energy transformation in the gas engine is exhibiting its best thermal performance while the expansion of steam in the engine cylinder is not carried to the most economic pressure limit.

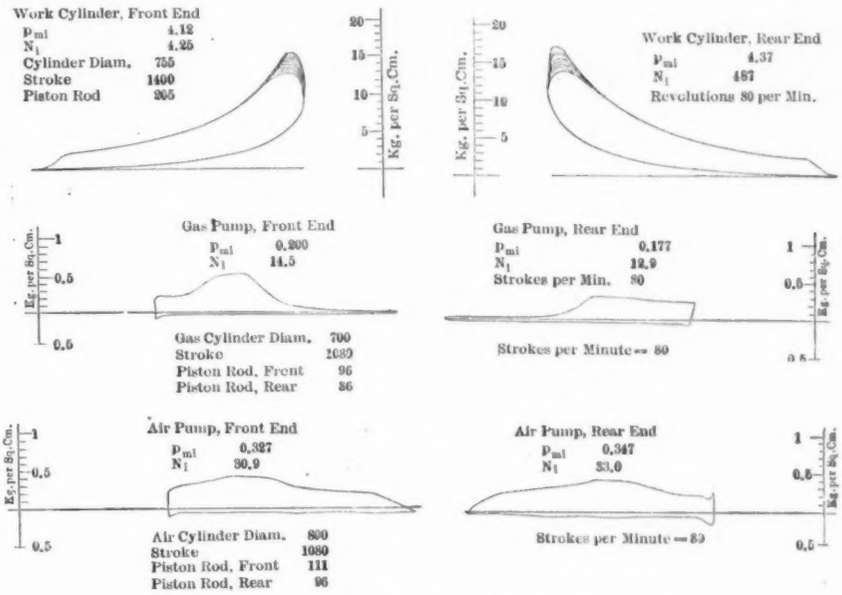


FIG. 109. DIAGRAMS FROM 600-H.P. KOERTING ENGINE

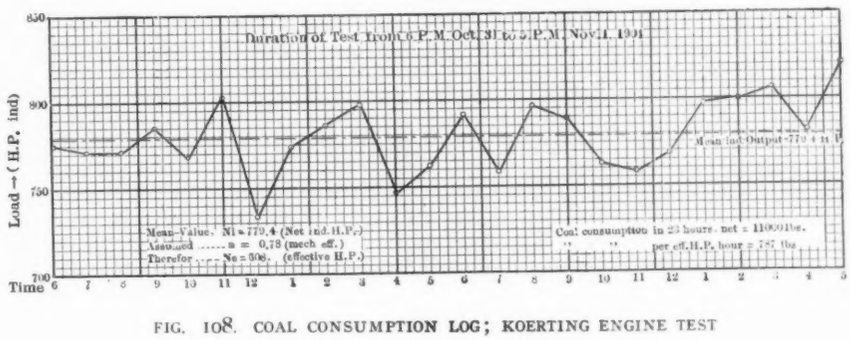

Duration of Test from 6 P.M. Oct. 31 to 5 P.M. Nov. 1 1901
Load → (H.P. ind)
Time
Mean-Value. Ni = 779.4 (Net ind. H.P.)
Assumed ..... n = 0,78 (mech eff.)
Therefor .... Ne = 608. (effective H.P.)
Coal consumption in 23 hours. net = 1100 lbs.
per eff. H.P. hour = 1.81 lbs

FIG. 108. COAL CONSUMPTION LOG; KOERTING ENGINE TEST

A comparison of performances obtained under load conditions which correspond to the rated capacity of the two types, or to any duty below the maximum, will put the gas engine at a disadvantage, since either compression is reduced or the calorific value of the mixture is impoverished, either of which will reduce the thermal efficiency. So it is only by comparing the results attained throughout the entire range of an identical load of the same seasonal, daily and hourly variations that the values recorded can be considered as something like definite. But here again it may be objected that identity of type, load factor, characteristics of particular application and class of service, the training of the operating staff, etc., are not alone sufficient; but that a difference in the geographical location of the respective plants, for example, will spoil the comparison, since it is known that the degree of altitude exercises an influence on the performance of gas engines which is indeed of no mean order of magnitude.

If we extend a comparative investigation to the total operating expenses, or still further, to the total cost of production per unit of power output, including fixed charges such as interest, depreciа-

tion, taxes, insurance, then conditions become even more complex, and a criterion by which the relative value of the two rival systems can ultimately be judged requires the consideration of so many variable factors that, with fairness to both types, we cannot claim the results obtained to be more than an approximation to the truth.

So to the critical student of the power problem emphatic statements such as are often to be found in catalogs of gas-engine manufacturers, sounding the death knell of the steam prime mover, seem rather ill placed. It is obviously better to convince the discriminating engineer and the public of the merits of our case by presenting the strongest argument of figures and facts, showing what extensive application this form of power generation has actually received, than to offer comparative tests of two rival systems built on an unscientific, and therefore weak and disputable basis.

If we consider that the world's total output of gas power has increased within the last four years in the ratio of 1 to 5.4, namely, from 181,000 horse-power generated in 327 gas engines, to 1,000,000 horse-power produced in large gas engines, of which one-half are "made in Germany," one-fourth in the United States, and the rest in other countries, then these figures will establish better than can any argumentation the fact that gas power has become a strong claimant for recognition in the field of power generation.

Volume and Pressure Delivered by Fan Blowers

A fan blower has the peculiar function of automatically increasing the volume of air delivered as the resistances are decreased, instead of always delivering a fixed volume of air regardless of requirements. On the other hand, if the blower be operated with a fairly free outlet in excess of its capacity area, and that free area be decreased, the pressure produced will immediately rise, thus tending at once to overcome the increased resistance. Therefore if a certain maximum pressure is known to be required the fan may be speeded to give this at such times as the conditions demand, while at other times, when less pressure or volume of air is required, proper manipulation of the blast gate will economize power.

Straight Line Expansion Diagrams

The paper presented by Hans Holzwarth to the American Society of Mechanical Engineers and abstracted on page 45 contains so much of which steam engineers, especially those who are trying to familiarize themselves with the turbine, should know, that we are led to smooth the way for those whose knowledge of mathematics and of graphics may not be adequate to a useful comprehension of the paper presented. Most engineers are familiar with the expansion curve and the manner of its laying out, but we will commence at the beginning.

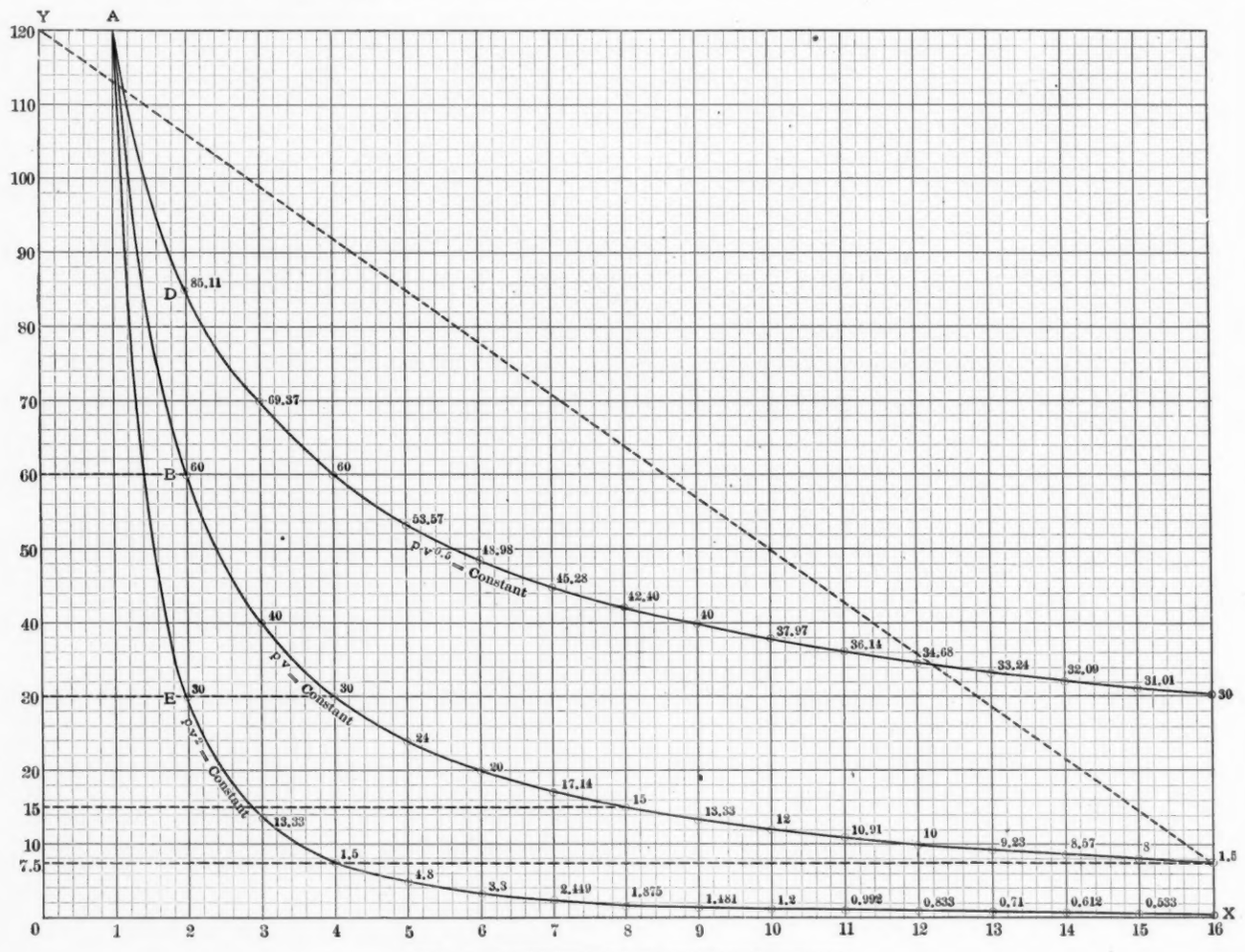


FIG. 1. CONSTRUCTION OF EXPANSION CURVES

In Fig. 1 let vertical distances or ordinates, as OY, represent absolute pressures and horizontal distances or abscissas, as OX, represent volumes.

When the temperature of a perfect gas is maintained constant during expansion its pressure falls in an inverse ratio to its volume; that is, if the volume is doubled the pressure will be halved; if the volume is increased to three times the original volume, the pressure will become one-third of the original pressure and so on. In this case the product of the volume and pressure remains constant.

$$p\ v = constant$$
$$p_1 v_1 = p_2 v_2 = p_3 v_3 \text{ etc.}$$

and for any particular volume the pressure will equal $p\,v$ divided by the given volume. For instance, the pressure when the volume becomes four times the initial volume (represented by $Y\,A$ or $O\,1$) would be:

$$p_4 = \frac{p\,v}{v_4} = \frac{120 \times 1}{4} = 30 \text{ lb.}$$

The pressures for the different volumes indicated on the $p\,v$ curve in Fig. 1 determined in this way are:

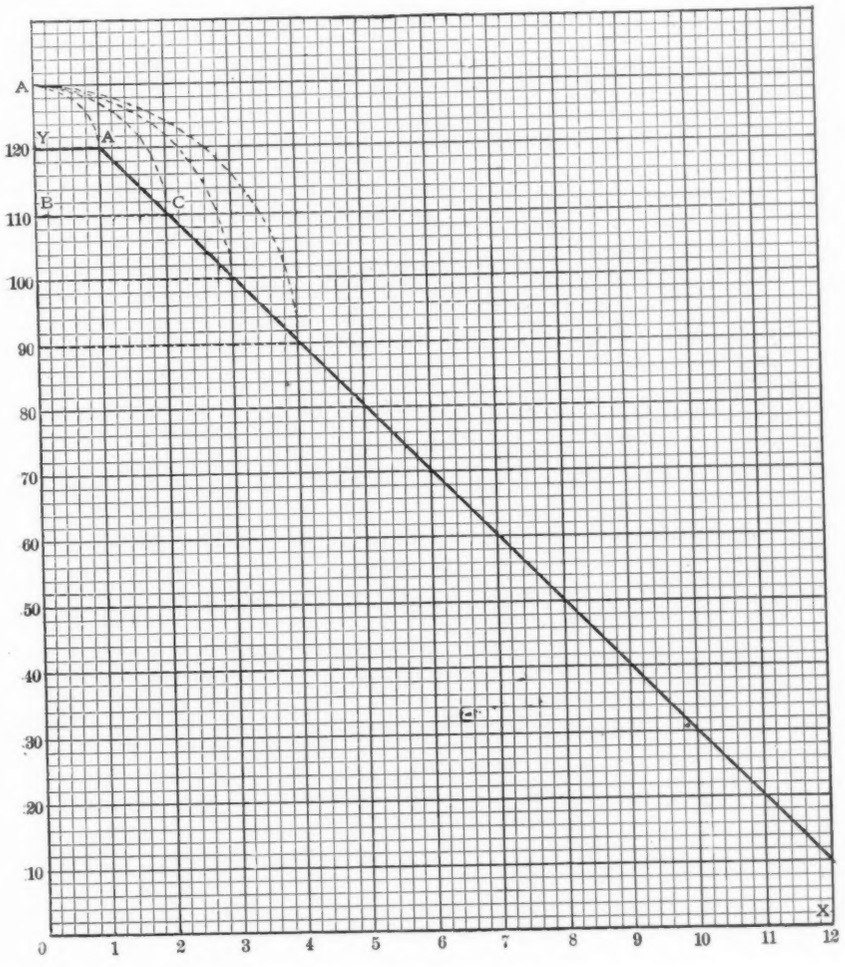


FIG. 2

Volumes.		Pressures.		pv.
1	×	120	=	120
2	×	60	=	120
3	×	40	=	120
4	×	30	=	120
5	×	24	=	120
6	×	20	=	120
7	×	17.14	=	120
8	×	15	=	120
9	×	13.33	=	120
10	×	12	=	120
11	×	10.91	=	120
12	×	10	=	120
13	×	9.23	=	120
14	×	8.57	=	120
15	×	8	=	120
16	×	7.5	=	120

In Fig. 1 the line $O\,Y$ indicates the zero of volumes and the line OX the zero of pressures. Set off to any convenient scale the distance $O\,1$, representing one volume, and upon the ordinate or vertical erected from this point at a hight corresponding to the given pressure, say for example 120 pounds, locate the point A. At double the distance from the zero line, indicating two volumes, erect another ordinate and upon it at B locate the pressure for that volume which would be:

$$p_2 = \frac{p_1\,v_1}{v_2} = \frac{120 \times 1}{2} = 60 \text{ lb.}$$

and in this way the points are located upon all the ordinates and the curve traced in.

This expansion at constant temperature is called "isothermal" expansion, and the $p\,v$ curve the "isothermal" curve. But the substance dealt with rarely retains the same temperature during expansion and is rarely or never a "perfect" gas. The pressures and volumes are likely to vary after another law and their relations, when plotted, to form a curve lying above or below that given by the $p\,v$ formula. In the upper curve when the volume is doubled the pressure, as indicated at D, is considerably greater than one-half of the original, while in the lower curve the pressure when the volume is doubled is, as indicated at E, one-quarter instead of one-half of the original. It is evident that the divisor (or the denominator v in the above fractions) must be less for the upper curve in order to make the quotient or pressure greater; and greater for the lower curve in order to make the pressure less. If this were done by multiplying v by a fixed quantity it would simply result in describing the $p\,v$ curve on a larger or smaller scale of volumes. It can be done, however, by using various powers of v.

Let us suppose for instance that:

$$p\,v^2 = constant$$

or:

$$p\,v^2 = p_2\,v_2^2 = p_3\,v_3^2, \text{ etc.}$$

Then pressure at any particular volume will equal $p_1 v_1^2$ divided by the square of the given volume. For instance the pressure when the volume becomes four times the initial volume represented by $Y\,A$ would be:

$$p_4 = \frac{p\,v^2}{v_4^2} = \frac{120 \times 1^2}{4^2} = \frac{120}{16} = 7.5 \text{ lb.}$$

The pressures for the different volumes indicated on the $p\,v^2$ curve of Fig. 1 are as follows:

Volumes.	Squares of Volumes.		$p = \frac{120}{v^2}$		pv^2
1	1	×	120	=	120
2	4	×	30	=	120
3	9	×	13.33	=	120
4	16	×	7.50	=	120
5	25	×	4.80	=	120
6	36	×	3.333	=	120
7	49	×	2.449	=	120
8	64	×	1.875	=	120
9	81	×	1.481	=	120
10	100	×	1.200	=	120
11	121	×	0.992	=	120
12	144	×	0.833	=	120
13	169	×	0.710	=	120
14	196	×	0.612	=	120
15	225	×	0.533	=	120
16	256	×	0.469	=	120

$$p_3 = \frac{p\,v^2}{v_3^2} = \frac{120}{3^2} = \frac{120}{9} = 13\tfrac{1}{3} \text{ lb.}$$

located at F. Other points may be located similarly and the curve traced in.

When a fraction is used as an exponent, as $v^{\frac{3}{2}}$, it means that v is to be raised to the power (i. e., multiplied by itself the number of times) indicated by the numerator of the fraction and that the root indicated by the denominator of the fraction is then to be extracted. Thus $v^{\frac{3}{2}}$ means that the cube root of the square of the volume is required, and if $v = 8$, $v^{\frac{3}{2}}$ would equal 4 because the square of 8 is:

$$8 \times 8 = 64$$

and the cube root of 64 is 4 for:

$$4 \times 4 \times 4 = 64$$

The fraction $\frac{1}{2}$ used as an exponent means that the square root is to be taken. For instance $v^{\frac{1}{2}}$ means that v is to be raised to the first power and the second or "square" root extracted. The first power is the quantity itself:

$$v^3 = v \times v \times v$$
$$v^2 = v \times v$$
$$v^1 = v$$

so that $v^{\frac{1}{2}}$ means the same thing as $\sqrt{v}$, simply that the square root is to be extracted. Since $\frac{1}{2} = 0.5$ this may equally well be expressed as $v^{0.5}$.

Now suppose the pressure varied inversely as the square root of the volume, the formula would be:

$$p\ v^{0.5} = constant$$

and

$$p_1 v_1^{0.5} = p_2 v_2^{0.5} = p_3 v_3^{0.5}, \text{etc.}$$

and the pressure p corresponding to any volume v would be:

$$p = \frac{p_1 v_1^{0.5}}{v^{0.5}}$$

To determine the pressure when the volume had been increased to four times the original in Fig. 1:

$$_4 = \frac{p\ v^{0.5}}{v_4^{0.5}}$$

As $v_1 = 1$ and any power or root of 1 is 1 this would be:

$$p_4 = \frac{120 \times 1}{\sqrt{4}} = \frac{120}{2} = 60$$

The pressures for the volumes indicated by the different abscissas are set off upon the ordinates and the curve traced in. These are as follows:

Volumes.	Square Root of Volumes.	$p = \frac{120}{v^{0.5}}$	$p\ v^{0.5}$
$v_1 = 1$	1	120	
$v_2 = 2$	1.41	$\times$ 85.11 =	120
$v_3 = 3$	1.73	$\times$ 69.37 =	120
$v_4 = 4$	2	$\times$ 60.00 =	120
$v_5 = 5$	2.24	$\times$ 53.57 =	120
$v_6 = 6$	2.45	$\times$ 48.98 =	120
$v_7 = 7$	2.65	$\times$ 45.28 =	120
$v_8 = 8$	2.83	$\times$ 42.40 =	120
$v_9 = 9$	3	$\times$ 40.00 =	120
$v_{10} = 10$	3.16	$\times$ 37.97 =	120
$v_{11} = 11$	3.32	$\times$ 36.14 =	120
$v_{12} = 12$	3.46	$\times$ 34.68 =	120
$v_{13} = 13$	3.61	$\times$ 33.24 =	120
$v_{14} = 14$	3.74	$\times$ 32.09 =	120
$v_{15} = 15$	3.87	$\times$ 31.01 =	120
$v_{16} = 16$	4	$\times$ 30.00 =	120

It is evident that if the exponent be made greater than 2 the curve will run below the lowest shown in Fig. 1; if the exponent be made less than 0.5 the curve will run above the highest in Fig. 1 and that by varying the exponent, an infinite number of curves may be put in between. For instance, the curve representing the adiabatic expansion of steam (that is, expansion during which it receives no heat from any source nor gives up any except that which is transformed into work) is $p\ v^{1.135}$. The exponent for adiabatic expansion of superheated steam is 1.3, etc.

It is natural that the reader should be wondering how the apparently complicated operation indicated by an irregular fractional exponent is to be performed. This would be difficult were it not for logarithms.

Arrange a series of numbers in geometrical progression, multiplying the value of each preceding place by some number, 2 for instance, as in the first column of the following table where each value is twice that above it. Opposite the 1 of this column place zero and continue the column in plain arithmetical progression.

Numbers	Logarithms
1	0
2	1
4	2
8	3
16	4
32	5
64	6
128	7
256	8
512	9
1024	10
2048	11
4096	12
8192	13
16384	14
32768	15
65536	16

It will be found that adding the logarithms corresponding to any two or more numbers will produce the logarithm of their product. For instance, multiply 32 by 8:

logarithm of $32 = 5$
logarithm of $8 = 3$
logarithm of $8 \times 32 = \overline{8}$

and opposite logarithm 8 is found 256 which is the product required.

What is the product of 4 x 128 x 64?

logarithm $4 = 2$
logarithm $128 = 7$
logarithm $64 = 6$
logarithm product $= \overline{15} = 32{,}768$

Suppose it is desired to find the fourth power of 8, that is:

$$8 \times 8 \times 8 \times 8$$

logarithm $8 = 3$
logarithm $8 = 3$
logarithm $8 = 3$
logarithm $8 = 3$
logarithm $8 \times 8 = \overline{12} = 4096$

But instead of adding we might better have simply multiplied the logarithm of the given number by the exponent of the power to which it is to be raised; thus:

logarithm $8 = 3$
exponent $= 4$
logarithm $8^4 = \overline{12} = 4096$

This becomes of importance with fractional exponents; for example, to find the value of $4^{4.5}$.

This is equivalent to

$$4^{4\frac{1}{2}} \text{ or } 4^{\frac{9}{2}}$$

i.e., the square root of the ninth power of 4; but it is more easily and directly found by multiplying by the exponent as follows:

logarithm $4 = 2$
exponent $= 4.5$
logarithm $4^{4.5} = 9 = 512$

Since evolution, the finding of roots, is the opposite process of involution, or raising to powers, the roots may be found by the opposite process of division by the index.

What is the square root of 1024?

logarithm $1024 = 10$
index of root $= 2$
$\overline{} = 5 =$ logarithm of root

and the number corresponding to logarithm 5 is 32, the root required.

This could have been done by multiplication by expressing the process of a fractional exponent:

What is the value of $1024^{0.5}$?

which is simply another way of asking what is the square root of 1024?

logarithm $1024 = 10$
exponent $= 0.5$
$\overline{}$
logarithm $1024^{0.5} = 5.0 = 32$ as before.

In the above system with 2 as a base the logarithms for the numbers between 16 and 32 for instance will be between 4 and 5. The tables ordinarily in use are made with 10 as a base thus:

Numbers.	Logarithms
1	0
10	1
100	2
1000	3
10000	4

so that the logarithms of the numbers between 10 and 100 consist of 1 and a decimal fraction, the numbers between 10 and 100 by 2 and a decimal fraction.

We have seen that a curve representing the changing relations of two factors whose *product* is constant will be a curve. If this be made the *sum* instead of the product the curve will become a straight line:

$$p \times v = constant = curve$$
$$p + v = constant = straight\ line$$

In Fig. 1 the area $YA10$ representing the product of the volume $O1$ and the pressure OY, is the same as that of the rectangle $60B20$ representing the product of the volume $O2$ and the pressure 60. In Fig. 2 instead of keeping this product constant we will keep the sum of the lines OY and YA constant so that $OY + YA = OB + BC$, etc., and the expansion path becomes a straight line instead of a curve. The difference between the lengths of the line OA and the length of a line representing a given pressure will be the length of a line representing the volume for that pressure, or the length of the volume line subtracted from OA gives the pressure for that volume.

Well, if we use the logarithms of the pressures and volumes instead of the natural values we shall get such a plus formula for:

$$\text{logarithm } p \times v = \text{log. } p + \text{log. } v.$$

If therefore instead of laying the diagram out with the natural values of the pressures and volumes as we did in Fig. 1, we lay it out with the logarithms of those values we shall get a diagram like Fig. 3.

The volume of one pound of steam at 120 pounds absolute is 3.67 cubic feet.

logarithm $120 = 2.08$, log. $3.67 = 0.56$
log. p + log. $v = 2.08 + 0.56 = 2.64$

Since the sum of the logarithms must be 2.64 we can find the logarithm for any pressure by subtracting the logarithm of the volume from 2.64 and find the pressures for the various ordinates of Fig. 3 thus:

p	log. p		log. v		
120	2.08	+	0.56	=	2.64
	2.04	+	0.60	=	2.64
	1.94	+	0.70	=	2.64
	1.84	+	0.80	=	2.64
	1.74	+	0.90	=	2.64
	1.64	+	1.00	=	2.64
	1.54	+	1.10	=	2.64
	1.44	+	1.20	=	2.64
	1.34	+	1.30	=	2.64
	1.24	+	1.40	=	2.64
	1.14	+	1.50	=	2.64
	1.04	+	1.60	=	2.64
	.94	+	1.70	=	2.64
	.84	+	1.80	=	2.64
	.74	+	1.90	=	2.64
	.64	+	2.00	=	2.64

Let v therefore equal 3.67. In calculations for the energy the pressure should be taken in pounds per square foot, but in laying out the diagram we can as well use the smaller value of pounds per square inch, as it will be proportional. Locating the point A 56 spaces from the zero axis horizontally and at a hight proportional to 2.08, the other pressures will come upon their respective ordinates as shown, but since they all lie upon the same straight line it is necessary only to locate one of them, as on ordinate 2.0, so as to get the direction of the line. Then by reading the hight at which the expansion line crosses a given ordinate we find the logarithm of the pressure and by looking it up in a table of logarithms or on the slide rule, the corresponding pressure. On the other hand by measuring the horizontal distance corresponding to the logarithm of any pressure we find the logarithm of the corresponding volume.

The area of Fig. 1 being proportional to the product of pressures and volumes represents energy and, of course, equal portions of the area represent equal amounts of energy. But, as will be seen from Fig. 1, the horizontal dotted lines dividing it into equal areas are at hights proportional to 7.5-, 15-, 30- and 60-pound pressure, mounting logarithmically, for 15 is twice 7.5; 30 twice 15, etc. By using the logarithms of the pressures as in Fig. 3 instead of the pressures themselves the logarithmic spacing of the dotted horizontals for equal amounts of energy becomes uniform and the amount of energy represented by any division of the diagram will be directly proportional to its hight and the diagram may easily be divided into stages of equal or desired amounts of energy each, and the range of pressures and volumes in each stage easily measured.

If the formula has an exponent as 1.135 for adiabatic expansion

$$p\ v^{1.135} = constant$$

$$\log.\ p + (1.135 \times \log.\ v) = constant$$

Since we wish to keep the starting point of the curve at A we will keep 2.64 for the constant and say:

$$\log.\ p + (1.135 \times \log.\ v) = 2.64$$

whence

$$\log.\ p = 2.64 - 1.135 \times \log.\ v$$

Instead of falling away an amount representing the logarithm of v the line falls off an amount representing 1.135 times that volume so that the vertical distance $a\ c$ is 1.135 times $a\ b$. The location of the line representing expansion with any exponent can be located by multiplying any tangent of the angle α (i. e., ab) by the given exponent and setting the pressure off on the vertical line which the tangent represents at that point, as at c, for adiabatic expansion of saturated and at d for superheated steam with an exponent of 1.3

Read Mr. Holzwarth's paper on page 45.

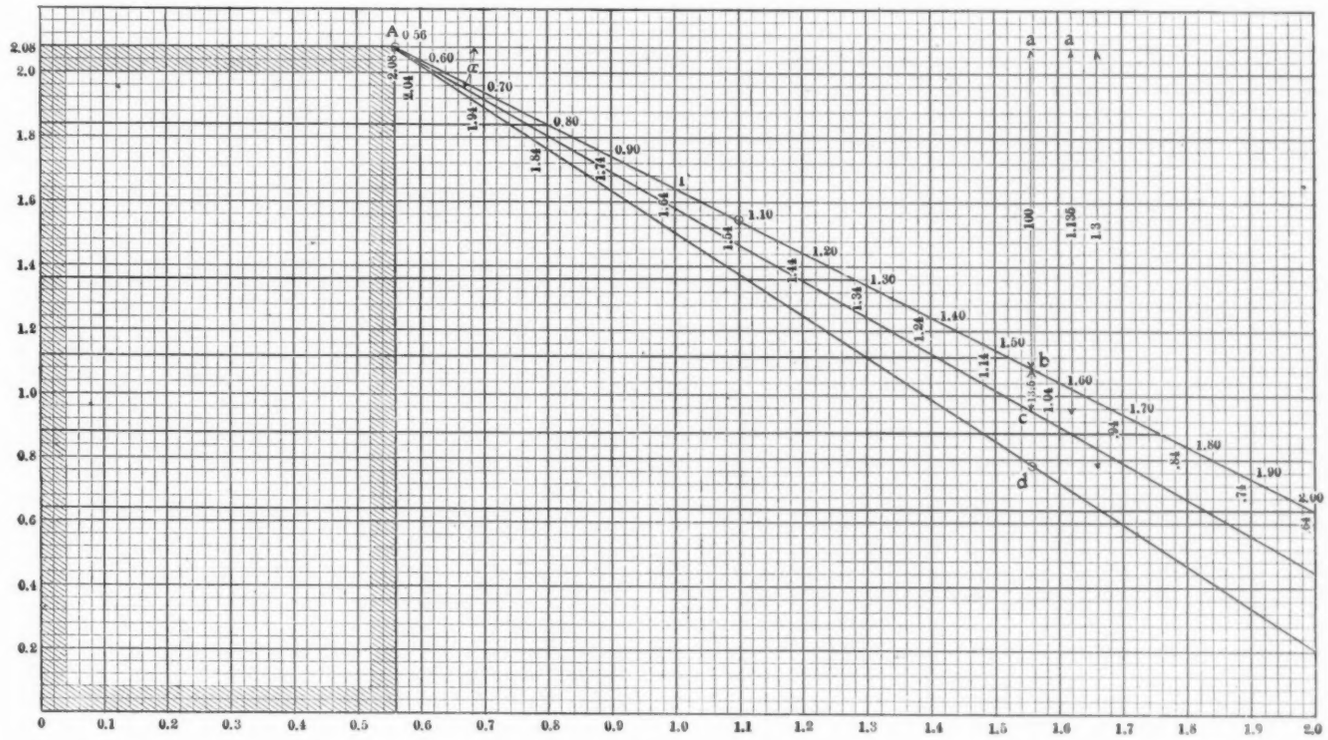


FIG. 3

Large Electrical Pumping Plant for Mexico City

The largest electric motors ever built for use in the Republic of Mexico are now being constructed at the works of the Westinghouse Electric & Manufacturing Company, for use in connection with Morris single-impeller centrifugal pumps to pump water from Xochimilco to Mexico City. Five pumping stations will be installed, and every precaution has been taken to prevent interruption of the service; a spare electrical unit and pump will be provided in each pumping station. In the first station will be placed three synchronous motors of 1200 horse-power each, to run at 720 revolutions per minute and pump 13,500 gallons of water per minute. Two motor-generator sets will supply the direct current for exciting the fields of the three motors. To reduce the supply-line voltage from 18,000 volts to that required by the motors, 3000 volts, nine 400-k.w. oil-insulated transformers, three to each motor, will be provided.

Station No. 2 will have two 60-horse-power motors pumping 4750 gallons of water per minute; station No. 3 will have two 160-horse-power motors pump-

ing 9500 gallons of water per minute, station No. 4 will have two 120-horse-power motors pumping 11,090 gallons of water per minute, and station No. 5 will have two 356-horse-power motors pumping 12,675 gallons of water per minute.

Power and Lighting Rates of the Jamestown Exposition

The rate for electric service at the Jamestown exposition, which will open at Hampton roads on April 26 of next year, are as follows:

RATES FOR LIGHT SERVICE.

100 kilowatt hours or less, 10 cts. per kilowatt hr
101 to 280 kilowatt hours, 9 cts. per kilowatt hr
281 to 625 kilowatt hours, 8 cts. per kilowatt hr.
626 to 1430 kilowatt hours, 7 cts. per kilowatt hr.
1431 and above kilowatt hrs., 6 cts. per kilowatt hr.

RATES FOR POWER SERVICE.

115 kilowatt hours or less, 9 cts. per kilowatt hr
116 to 360 kilowatt hours, 7 cts. per kilowatt hr.
361 to 770 kilowatt hours, 6½ cts. per kilowatt hr.
771 to 1365 kilowatt hours, 5½ cts. per kilowatt hr.
1366 to 2225 kilowatt hours, 4½ cts. per kilowatt hr
2226 and above kilowatt hrs., 4 cts. per kilowatt hr

The exposition company will make all necessary connections from the exposition service main to the main switch and fuses installed by the consumer, together with all necessary meters and apparatus for measuring the current consumed, at the consumer's expense. Alternating current at a frequency of 60 cycles and at approximately 110 volts will be furnished for lighting purposes on the entire ground. On the War-path direct current of the same voltage will be also available. For the power service direct current at approximately 500 volts will be furnished, but special arrangements may be made with the department of electricity for furnishing alternating current for small motors.

The rates above given are subject to a discount of ten per cent. for prompt payment, and are, of course, subject to the terms of any special contract which a consumer may execute.

Experiments initiated by the Manchester Steam Users Association are in progress at the National Physical Laboratory to determine the specific heat of superheated steam. The method employed is to superheat the steam by means of electrically heated wires, the amount of heat being determined by electrical measurement. Extensive experiments in the same line, but with other methods, are being conducted at Sibley College and at Charlottenburg. It will be interesting to see how the results compare.

The vertical Riedler pumping engine designed by Henry A. Allen for the Pennsylvania Water Company has been tested by Charles Hague, M. E., of New York, and showed the record duty of 157,349,000 foot-pounds per 1000 pounds of dry steam.

ELEMENTARY LECTURES ON ELECTRICAL ENGINEERING—XII

When the armature of a dynamo or motor is revolving and the field magnet is excited (magnetized) any given part of the armature core undergoes four changes in its magnetic condition during its passage in front of one pair of field-magnet poles. Referring to Fig. 31, it will be evident that the direction of the magnetic flux through the part *a* of the armature core inclosed within the dotted-line square is from the core periphery toward the center, this part of the core being at the instant under a north magnet pole. When the core has turned through one-fourth of a revolution to the position illustrated in Fig. 32, the flux has been reversed through the part *a*; during the passage of the part *a* from the north to the south pole, it has been demagnetized, of course, there being no source of magnetic flux between the two poles. Following it in its source through a complete revolution it will be found that when it has got back to the position in Fig. 31, it will have been magnetized four times, in alternate directions, and demagnetized four times, making eight changes that have occurred in the one revolution, or four for each pair of magnet poles that it has passed.

The rapid magnetization, demagnetization and reversal of the iron in the core is resisted by the iron and power is required to enforce these changes. The resistance to magnetic changes is called "hysteresis." In order to reduce the "hysteresis" effect as far as possible, the iron or steel of an armature core is made just

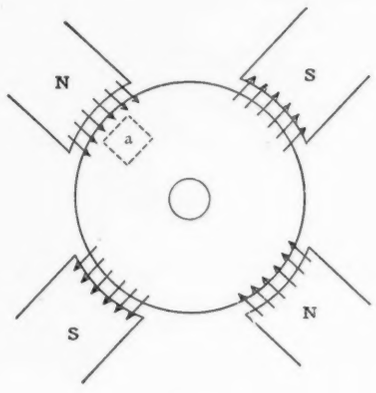


FIG. 31

as soft as possible, soft iron or steel responding more readily to magnetic influence than hard iron or steel. The loss due to hysteresis is proportional to the rapidity of magnetic change and increases with increasing magnetic density, through not in direct proportion; the loss by hysteresis increases more rapidly than the increase in magnetic density, but not as rapidly as the square of the density. According to Dr. Steinmetz, the hysteresis is proportional to the 1.6 power of the magnetic density.

There is one other cause of loss in the armature core, namely, the induction by the field magnetism of "eddy" currents in the iron. If the motion of the armature past the field-magnet poles induces an electromotive force in an armature conductor lying parallel to the shaft, it will also induce an electromotive force in the iron of the core parallel to the shaft, if the iron is solid axially. This electromotive force will, of course, cause currents

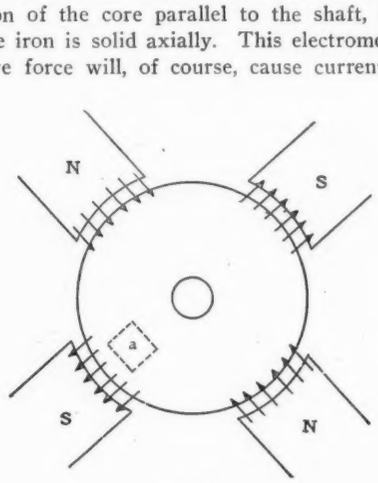


FIG. 32

to flow in the closed circuit offered by the armature core near its periphery and the shaft through its center. This is illustrated roughly by Fig. 33. If that part of the core surface lying between the heavy dotted lines is passing beneath a north pole of the field magnet and its direction of rotation is that indicated by the arrow *r*, then an electromotive force will be induced in the iron at the surface which will cause currents to flow from the front to the back end of the core as indicated by the group of arrows. At the back end the currents will flow across the end to the center, back through the center and across the front end to the periphery again, forming a closed circuit. These currents, of course, do no useful work and represent a waste of energy; moreover, they exert an objectionable reaction upon the field-magnet poles.

In order to reduce the generation of eddy currents, the armature core is built up of very thin disks of iron or steel instead of being solid from end to end. The disks are coated on both sides with an insulating compound to prevent the flow of currents from disk to disk. Even these precautions are not wholly preventive; eddy currents are induced in each disk, but they are relatively insignificant. Sometimes, however, the edges of the disks become burred and touch each other; then eddy currents flow along the core surface from end to end, or as far as the burred edges are in contact, and cause excessive heating. The loss by eddy

currents is proportional to the square of the speed, the square of the number of field-magnet poles, the square of the magnetic density in the iron and the square of the thickness of the disks.

From the foregoing statements it will be evident that the losses due to hysteresis and eddy currents in the armature core of an existing dynamo or motor will not depend on the load carried by the machine. If its speed is constant, the core losses will depend on the field strength only.

There is one more armature loss, and this is directly proportional to the load. It is the loss in the armature winding, due to the resistance of the conductors. The loss in any electrical conductor or group of conductors is proportional to the resistance and to the square of the current. For example, if a conductor or group of conductors has a resistance of 5 ohms and a current of 10 amperes be passed through it, the drop in volts will be $5 \times 10 = 50$. Since watts are the product of volts and amperes, the loss in this case will be $50 \times 10 = 500$ watts. This

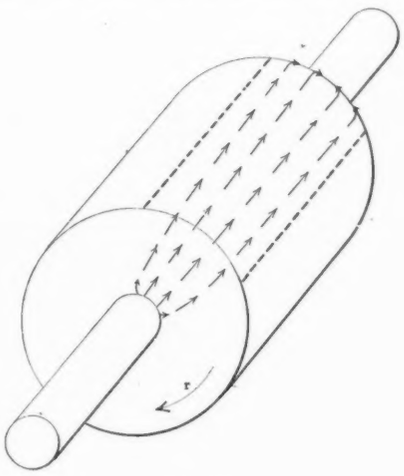


FIG. 33

is obviously the same as multiplying the resistance by the square of the current, because it was multiplied by the current once to get volts, and the volts were multiplied by the current to get watts.

All losses of energy are manifested by the heating of the body in which the losses occur. The loss in an armature core due to hysteresis and eddy currents heats the iron or steel of the core; the loss in the winding due to its resistance heats the copper wires of which it is composed; the loss by friction in a bearing heats the surfaces that rub against each other, and so on. Of course, the heat produced by these losses is not confined to the parts in which it originates; it is transmitted rapidly to adjacent parts of the machine.

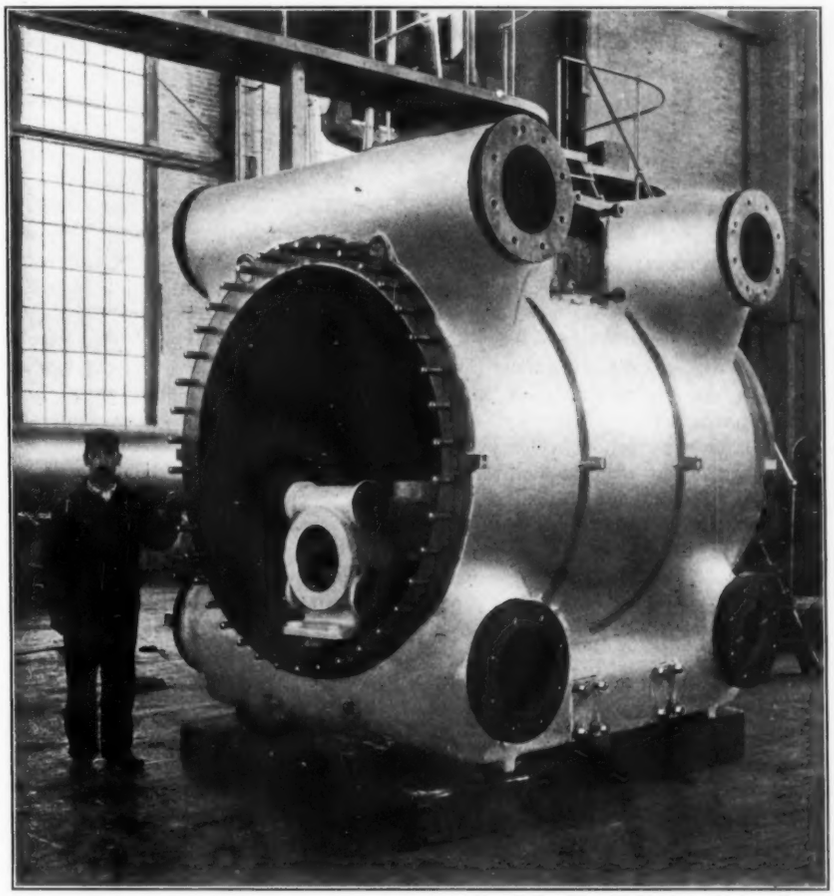
The loss in the armature winding and brushes may be readily ascertained by running the machine until it warms up to its normal running temperature, then shutting it down and passing a small current through the armature alone and measuring the volts at the brushes while this current is passing. Dividing the volts at the brushes by the current passing through the stationary armature will give the total resistance of the winding and the brushes; multiplying this by the square of the full-load armature current will give the full-load armature loss due to resistance.

The losses in the armature core alone cannot be very easily ascertained, but the combined losses due to hysteresis, eddy currents and friction in the bearings can be tested out without much difficulty if there is available a source of current of a voltage equal to that of the machine to be tested. The process is as follows: Run the machine as a motor at its rated speed without any load and measure the current passing through the armature and the voltage at the brushes. Multiply the current by the voltage and the result will be "total no-load watts." Stop the machine and pass through the stationary armature a current equal to that which was passing when it ran without load; measure the volts at the brushes while this current is passing and multiply them by the current; the result will be the loss by resistance while running at no load, and if this be subtracted from the "total no-load watts" the final result will be the watts lost by hysteresis, eddy currents and friction combined.

Tiny and Titan

The smaller of the two cylinders shown in the accompanying engraving is an 11x36, being the high-pressure cylinder of a cross-compound condensing engine direct connected to a generator now running in the shops of its builder, the Wisconsin Engine Company, of Corliss, Wis.

The larger cylinder, in the bore of which it sets, is a 64x54, the low pressure of a 2500 horse-power vertical cross-compound engine of which six have been furnished by the Wisconsin Engine Company to the Public Service Corporation of New Jersey for their station at Newark.



TINY AND TITAN

The United States Civil Service Commission will conduct examinations on January 9-10-11 to secure eligibles from which to fill vacancies in the position of architectural and structural steel draftsman at $125 per month, in the office of the engineer of the Twelfth Light-House District at San Francisco, and to fill at least ten vacancies in the position of mechanical draftsman at from $1000 to $1200 per annum at various stations. Application to the Civil Service Commission at Washington, D. C., for form 1312 will produce the necessary information.

CORRESPONDENCE AND DISCUSSION

Correspondence upon Topics of Interest Relating to Power Is Especially Solicited, and Accepted Contributions Are Paid For—Answer Early in Month if Matter Refers to Preceding Issue

Export Mistakes

I read with interest your article on this subject in the July issue, because we were the unfortunate purchasers of the boilers referred to therein.

The exception which the manufacturer of these boilers takes to the story, as set forth in your September issue, leads me to make the following remarks:

These boilers were ordered on December 1, 1898, and delivery was guaranteed for April 22, 1899. The boilers were not delivered until July, 1899, and in fact the particular boilers on this order were not even delivered then but, as various companies belonging to the same group had placed orders for identical boilers, we were enabled to take some over from a neighboring mine, as we were in urgent need of the extra boiler capacity, and when our boilers did eventually arrive they were delivered to that mine in place of those borrowed.

On starting the erection, which was carried out by the firm that supplied the boilers, the first thing that appeared was that the radius of the neck of the tube headers was not the same as the radius of the drums, by such an amount that it was necessary to heat the flanges of the tube headers and bend them until they would fit.

Another serious defect that was discovered was that at the junction of the plates in the drum shells the corner of that particular plate which came between the other two had not been scarfed down, and it is a positive fact that at this point the calking at the outside of the seams was the only thing depended upon to make the boiler tight.

If this is the sort of work that would be supplied to your home market, I am sorry for the purchasers.

Sundry other defects developed, which I will not trouble you with, further than to state that I still have specimens of rivets that were cut out from the drums, which I am sure the manufacturer of these boilers would not care to have exhibited. If you should care for further details, I would refer you to the gentleman who was sent out here by the boiler manufacturer to make a report on the condition of the boilers, and who arrived full of the same ideas about the boilers being in every respect the same as would have been made for the home market, having passed shop tests, etc. After a careful inspection of the boilers, he agreed that we were perfectly justified in condemning them.

With regard to the statement that it was a political or industrial condition which led to the rejection of the boilers, I can only give this a most unqualified denial, and the statement that boilers from a German firm and from an American firm of exceptional reputation were rejected at the same time and by the same final purchaser for any reason whatever is entirely untrue, as this company has never rejected any other boilers in the course of its existence, has never purchased any German boilers, and never any American boilers other than those under discussion. We now have in use some four thousand horse-power of boilers made in England and Scotland.

In view of the facts of the case, the attempt of the makers to repudiate your original statement, which was essentially correct, is unjustifiable (to use no harsher term), but quite on a par with the quality of the boilers they supplied us.

I might add that, in conformity with local practice, thirty per cent. of the value of the contract was paid to the local agents on shipment, and a further twenty per cent. upon arrival of the boilers on the property, a total of half the contract price, for which sum (together with about half as much again, in the way of expenses incident to the partial erection and removal of the boilers) we still hold a judgment awarded by the courts against the unfortunate local agents of the makers of the boilers. The amount in question has been written off our books as a bad debt.

A. W. Peirce,
General Manager, General Electric Power Co., Ltd.
Germiston, Transvaal.

A. J. A. Replies to Mr. Wakeman's Criticisms

In the December number W. H. Wakeman expresses amusement at my so called turning about of his question No 1. I am glad to furnish amusement to anyone who is able to find it in an honest effort to analyze an unnecessarily obscure statement of a simple proposition. The fact that Mr. Wakeman thinks an "automatic cut-off" engine is not a "variable cut-off" engine is no excuse for the balance of us to indulge in such hallucinations.

I used 15 pounds for the atmospheric pressure in questions Nos. 4 and 6 because these questions specifically stated that this pressure must be used and despite the fact that I knew it was wrong. Unless it be at the bottom of a mine shaft, there is no such atmospheric pressure. Question 5 read: "Is the terminal pressure above or below atmospheric pressure when the barometer stands at 29.9 inches?" Any sane man would suppose that an examiner priding himself on his accuracy would have found it fairer to say, "Is the terminal pressure above or below the atmospheric pressure, the latter being taken at 15 pounds?" If the barometer reading was to be neglected in favor of 15 pounds atmospheric pressure, why was the barometer mentioned, and why does Mr. Wakeman whip the devil around the stump and claim the answer to be faulty because it uses the data given instead of something else? Moreover, I cannot believe that readers of Power, or any other ambitious engineers will be discouraged by decimal fractions carried to two places.

It is hardly worth while to consider Mr. Wakeman's criticism of my answer to No. 10. The definition of a boiler horse-power is given in the answer and readers of Power do not need that I or Mr. Wakeman should state that an engine horse-power is equivalent to 33,000 foot-pounds per minute. Neither do they need to be told that this rate of work is not necessarily equivalent to the work represented by a boiler horse-power. It may be pointed out, however, that the engine mentioned in the question might easily consume as much or more steam per horse-power than that which corresponds to a boiler horse-power. Thus in a table on page 753 of his Pocket-book, Kent shows that such an engine may consume from 24 to 32 pounds per hour per horse-power at 80 pounds pressure. For the 70-pound pressure of the question the consumption would be still more owing to greater condensation. Thus it turns out that each horse-power of the engine eats up the steam furnished for each horse-power of the boiler and the two *happen* in this case to represent about the same thing. It would have to be a larger shop than could be driven by a 25-horse-power engine to utilize the

87 — 25 = 62 boiler horse-power in heating. Hence it is plain to anyone except Mr. Wakeman that a big leak somewhere is indicated. Possibly it may be a poor heating system but more likely it is in the engine and auxiliaries as stated in the answer given. As to the "auxiliaries generally used with a 25-horse-power engine" it would require only one good husky steam-boiler feed pump to make a big increase in the steam used. Perhaps Mr. Wakeman will concede that such a pump would be needed.

In my answer No. 12 where I stated that the blow-off pipe should be protected by masonry, Mr. Wakeman, reading backward as usual, seems to infer that I advocate grouting the pipe in a mass of concrete as if it were the foundation bolt for an engine. Where the pipe is tapped into the bottom of the boiler, as is the general practice, it is considered better to put a brick protection in front of the pipe rather than rely solely on an asbestos-rope covering.

Regarding answer 13, it is quite usual, I believe, for engineers to refer to the amperes as being an index of the load when the voltage of the circuit is stated; they also know that when the volts are constant the amperes are an actual measure of the load. I did overlook the fact that the motor was connected between the outside and a neutral wire. Probably the reason I did this was that my mind was so strongly impressed with the correct way of connecting the motor that I lost sight of the obsolete method stipulated by Mr. Wakeman. A. J. A.

Chicago, Ill.

Flues and Tubes—Sacrificing the Steam Line to Ease in Shutting Down

I am not a methodical reader of POWER for want of time but there are several things in August, 1906, number which I have just picked up which excite me to suggest and to inquire.

W. H. Wakeman on page 486 takes the fraternity to task for the indiscriminate use of the terms "flue" and "tubular" as applied to boilers, and winds up with the statement that a flue of less than four inches diameter is a tube, and a tube of more than four inches is flue, or words to that effect, and makes no other distinction.

I have always thought, for what particular reason I can't say, that in a flue the gases of combustion circulated inside, and the water outside, while in a tube this order is reversed. Is Mr. Wakeman's statement the correct and only one?

W. J. T's case of the engineer who governs his Corliss engine with the throttle is, I think, rather extreme, but I have always liked to run with the throttle open only a half or a full turn more than is necessary to give full power. In case of accident it takes just that much less time to stop. GEO. MACK.

Elliston, Montana.

A Question of Belt Operation

In reply to the question of belt operation by R. T. Strohm in the November issue (page 695) I would say that the belt would run to the "low" side of the pulley in the figure.

There is a fallacious notion prevalent among engineers that a belt always "runs to the high side of a pulley." This sometimes happens, but it is only when the high side of the pulley is the point to come first in contact with the belt, for it is an invariable rule that a belt always leads to the first point of contact.

When a pulley is crowned the middle of the belt will tend to run on the high point of the crown, provided the pulleys are in line, not because it is the high point, but because the belt first comes in contact with the pulley in the center of the crown.

In Mr. Strohm's problem the belt would also tend to lead off the driving pulley *F* toward the right, as the belt would be returning to it at an angle caused by the tilt of the pulley *G*.

This principle is very well illustrated in the running of wide band-saws (which may be regarded as steel belts; indeed when first introduced they were called belt-saws as frequently as band-saws) where the lower pulley is placed level and stationary while the upper has an adjustable tilting device and is tilted just far enough forward to cause the teeth to run clear of the wheels. It was formerly the custom to grind the wheels with a slight crown but this has been generally discontinued as it was found that the upper wheel required more tilt and consequently caused a greater strain on the back edge of the saw.

Almost every engineer has made use of this principle in moving machinery on rolls and if any wish to verify it they may do so by placing a stick of timber on a pair of rolls as shown by the sketch herewith (Fig. 2); it will be found that the timber rolls in the direction of the arrows. ALVIN R. CAMPBELL.

Astoria, Ore.

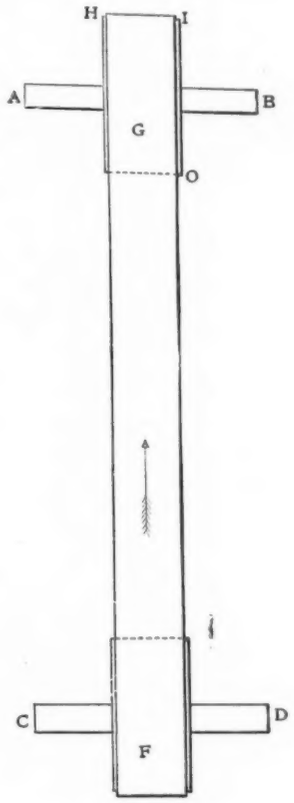


FIG. 1

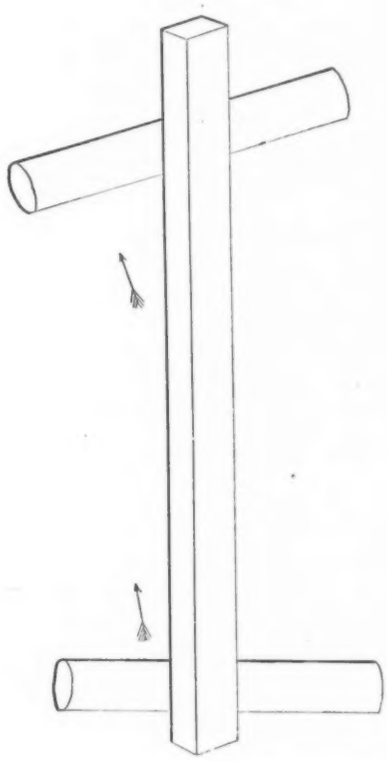


FIG. 2

A QUESTION IN BELT OPERATION

Mr. Strohm's belt will certainly run off on the side which he calls "low." His sketch shows the shafts not parallel, but with the edges of the pulleys in the same plane, which is an impossibility, if the pulleys run true. By an inspection of the accompanying sketch (Fig. 3) it will be readily seen that the belt will run off at the side marked *A* as soon as started up, whether the angularity is much or little.

Crown-faced pulleys are designed to remedy this if the shafts are but slightly "out." This question of "high side" and "low side" is a very old one and is due

to a misapprehension of other and well known facts. In locating pulleys for belting, whether straight, crossed, quarter turn, or any other of the many ways in which belts are run, the thing to fix in mind is that the belt on the "running on" side of the pulley must be parallel with the direction of motion of the pulley. The *running off* side will take care of itself, whatever the angle may be. The other part of the proposition is that if by "high side" we mean a "crown" or other enlargement of the diameter of the pulley, whether in the middle of the face or at one edge, the belt will quickly find it and seem to cling to that particular point. The reason is that the surface velocity is greater at that point and consequently by stretching the belt causes a slight curvature which would lead it back should it try to leave that point. In the other sketch (Fig. 4) I have endeavored to make it more clear. Should the belt start toward the side marked *A*, the greater velocity at the other edge would cause

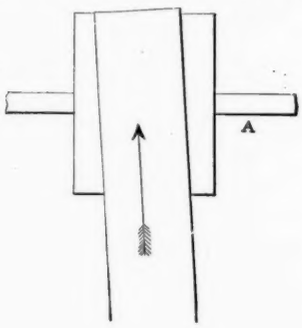


FIG. 3

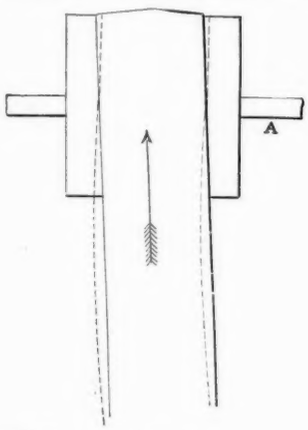


FIG. 4

A QUESTION IN BELT OPERATION

the belt to assume the position indicated by the dotted lines, and this would immediately lead it in the other direction. The reader has probably noticed that a belt that has been stretched at one edge by rubbing against some stationary object or otherwise, will always tend to run off the pulley at that side.

This is my explanation. If I am wrong, should be glad to be corrected.

A. W. BLANCHARD.

North Adams, Mass.

In the November number of POWER Mr. Strohm wants to know which side of the pulley *G* the belt will run off. A little consideration will show and experience proves that it will run off the "low" side *I*. However, if one side of the pulley were of greater diameter than the other and the shafts were parallel the belt would run to the "high" side, the greater speed of that side pulling harder on the belt.

W. RUSSELL COOPER.

Greencastle, Ind.

[The foregoing letters were received before the December number was printed, but too late to appear in it.—EDITORS.]

Comments

The method of running the feed-water pipe on the interior of a boiler is doubtless the most satisfactory method that is in use at the present. It is what is regularly recommended by the Hartford Insurance Company and is in use in thousands of boilers. As Mr. Howard says: there are many engineers who would not adopt any other method of feeding a boiler; those who oppose the top feed are not acquainted with its merits.

I do not think that a brass bushing is the thing to put into a boiler shell as the expansion between steel and brass is considerably different. For instance, brass will expand 0.0000106 of an inch for every degree Fahrenheit: steel will expand 0.0000066, or a difference of 0.000004 of an inch; this is for a length of one inch. Just what the expansion of his bushing would be would depend upon the steam pressure carried. Supposing that it were 100 pounds, then the temperature would be 327.6 degrees, which would amount to:

$327.6 \times 0.000004 = 0.0013$ of an inch.

Just what effect this difference in expansion would make in the boiler shell may be questioned, but it is enough to make the difference between a tight and an easy fit and to cause a leak under some circumstances; it would be better to connect an iron pipe direct to the shell of the boiler.

Referring to the question of John I. Baker regarding running Corliss engines under as well as over, I would say that there is no more objection to running a Corliss engine under than there is to running any other engine under. No engineer has any desire to operate an engine running under; the objection is that the crosshead gives trouble by pounding unless very close adjustments are made on the crosshead shoes. If very close adjustments are made there is some danger that they will heat up after the engine is in operation in case the temperature of the room becomes higher than it was when the adjustments were made. If the engine is working hard the space between the shoes and the guides should not be more than the thickness of a piece of ordinary writing paper. If the engine runs over there is not so much need of close adjustment; in fact I have seen engines operating with a space between the top shoe and the guide that one could easily see through.

I have been taught that a belt will always run to the tighter or higher side of a pulley. In the case cited by Mr. Strohm, the rim of the pulley at *H* will be the tighter or higher side and, therefore, the belt should run to that side of the pulley when the belt is running in the direction indicated by the arrow.

Cleveland, Ohio. E. P.

"How One City Plant is Run"

As an engineer I do not hold a brief for either party, but should like to make some comments on the article by "Subscriber" from Perry, Oklahoma, on page 700 of your November issue, entitled "How One City Plant is Run." Without giving any data as to the capacity of the engines, auxiliary pumps and machinery, sizes of boilers and general lay-out of the plant, he finds fault in his opening sentence with the boiler-feed pumps and connections; but possibly if he had taken the trouble to look deeper he might have found the reason why the boiler-feed pump was in the engine room and have found that the increased friction due to a little more pipe or a few more elbows was inconsiderable beside some advantage which he fails to find. I surmise the reason but do not advance one, having no knowledge of the conditions.

The connection from the exhaust pipe to the heater appears to be rather inadequate, but perhaps the drawing is not made to scale.

There is one sentence in this remarkable letter that I commend to the attention of your readers particularly: "Each pump had a gate valve on its exhaust pipe close up to the pump. I did not see much use for them, and I think they are entirely out of place, for in big pumps it is easier to overcome a slight defect by throttling the exhaust than by remedying the defect, but at the expense of the coal pile."

My experience is that where you have a main engine and a number of auxiliaries exhausting into a common exhaust pipe, a valve on the exhaust pipe and close up to each pump is not only in place, but an absolute necessity. Otherwise if an auxiliary breaks down, and the engineer wants to examine it, how is he to be able to do so unless he can isolate it by closing off the exhaust. As to throttling the exhaust, I fail to see what good this would do in remedying defects, or what its consequent expense on the coal pile might be.

In reference to discarding the old boilers and installing new ones, so many con-

siderations come up in many cases that it has been found true economy to take out a brand new boiler and replace it with another having less grate and more heating surface. Every case in this particular ought to be judged on its own merits, the condition and general suitability for the work in view being known.

Seattle, Wash. J. GREEN.

"Cheaper Power"

I was much interested in reading the letter by E. N. Percy in the November number of POWER. The problem of getting cheaper power from some source has, I believe, come into the mind of almost every person of a mechanical bent at some time or other. I have often thought of the wave proposition and here in Chicago as I pass along the lake front every morning there is hardly a day but what this comes to mind. To see a huge wave dash along the pile breakwater and hurl itself into spray is certainly enough to make one think of the forces of nature that have not yet been tamed. No one knows when it may be found commercially economical to harness such forces as this. When in Portugal at one time I saw brushwood burned to generate steam for railway trains and from this one would think that the time for cheaper power had about arrived for at least one country.

Windmills could undoubtedly be put to good use on many farms to shorten work by using the power to drive machinery. It is on farms and in open country that the windmill has its greatest value but there are too many farmers who have not the necessary capital to buy the mill and the machinery in the first place, so that they must go without the conveniences of the cheaper power. Then there is the non-mechanical mind of the farmer that would make it difficult for him to keep the machinery in order.

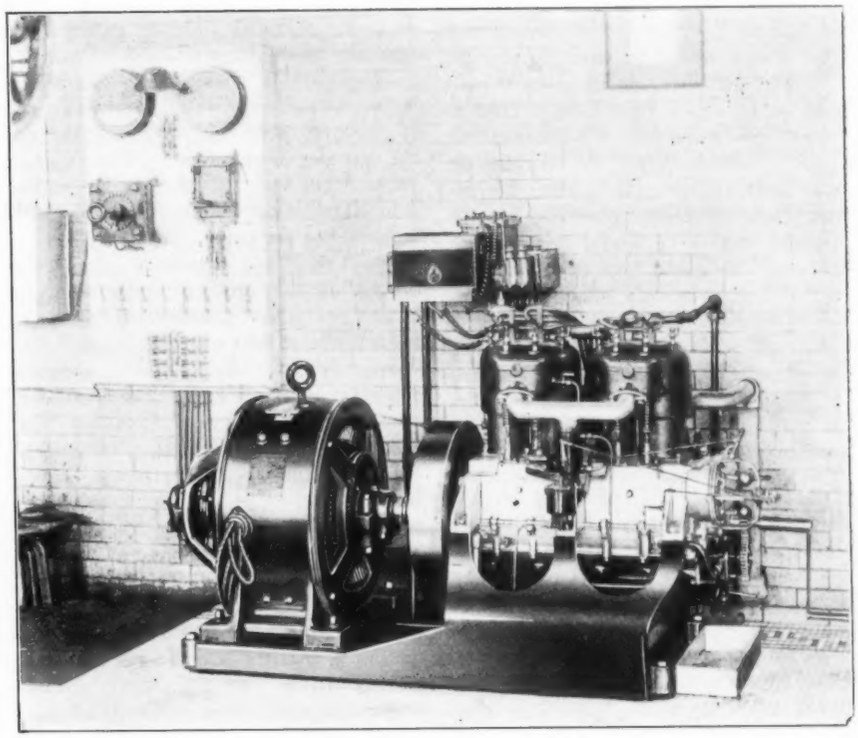
I think the limits of space available would hardly permit the sun-heated boiler to come into general use but the idea of ammonia pipes in a desert seems to have its advantages for that kind of country. It will be necessary to get some means of making the joints tight so that they will stay tight and not break when the temperature of the pipe system changes. It will also be necessary to have an expert to plan the piping layout so that the expansion can be cared for. There must be no air leaks into the system and the ammonia must not become too hot or it will decompose and eat up the pipes. The idea of power from expanding rods as in the clock is good. A short rod expanding a little distance exerts a great power and this could be put to use by magnifying the motion with a system of levers.

Of all Mr. Percy's ideas I think the one for catching the energy in the tides is the best. I see no difficulty in the way of the tower plan except the excessive first cost of the installation. This is, indeed, the trouble with all the so-called cheap powers and particularly with water power. It may be quite possible, however, to find a location where the conditions would justify the cost of the tide-lock towers. E. S. HAWKINS.

Chicago, Ill.

Application of an Automobile Engine to Dynamo Drive

The accompanying illustration shows an adaptation of a high-speed automobile gasolene motor to the work of dynamo driving. The dynamo is a 10-kilowatt Westinghouse generator, rated at 40 amperes and 225 volts, and it runs at 800 revolutions per minute. It is used in the factory of the Packard Motor Car Company, of Detroit, Mich., to supply electric light throughout the office and factory at times when the main power plant is not running, as at noon and in the evening. Nearly all of its running time it carries the full load of 40 amperes.



AUTOMOBILE ENGINE DRIVING INCANDESCENT LIGHTING MACHINE

The generator is direct-connected by a rawhide-strap flexible coupling of special design to a Packard marine motor, which, with the exception of such details as crank-case construction, exhaust piping, etc., is the regular 1906 Packard automobile motor, rated at 24 horse-power on the French basis. The four vertical cylinders are of 4½-inch bore by 5½-inch stroke, and are cast in pairs with the water-cooling jackets and valve chambers integral. The valves are all mechanically operated and interchangeable. The exhausts are on one side and the inlets on the other.

The ignition is by jump spark, the current being obtained from a low-tension Eiseman magneto and transformed in a single non-vibrating coil. There is a supplementary battery and vibrating coil for emergency. Lubrication is by splash from the crank case, with the oil forced to the case through glycerine sight feeds by a plunger pump, which is direct-driven by the engine. All of these features are exactly like the car motor, the appurtenances of the latter being used throughout without alteration.

Regularity of running is obtained by a centrifugal governor on the inlet-valve cam shaft and operating upon the butterfly throttle in the carbureter. The steadiness with which the engine runs under this throttle is shown by the voltmeter when a test bank of 160 incandescents in the engine room is thrown in and out suddenly. From no load to full load of 40 amperes there is not a variation of five volts. In fact, the unit is so satisfactory as an auxiliary generator that the chief engineer, accustomed all his life to steam plants, is particularly fond of it and says that he gives it less attention than any engine of any kind which he ever handled.

Detroit, Mich. E. RALPH ESTEP.

"Should a Corliss Engine Run 'Under'?"

Mr. Baker's question in the November issue as to an engine running "under" attracted my attention, as I am in charge of a Fitchburg compound-condensing engine which runs "under." The worst fault I find with it is that it throws oil from the crank and connecting rod all over the engine frame. Another objection might be that as most of the wear would come on the under side of the top crosshead guide, it would be slightly more difficult to give it proper lubrication. A. W. BLANCHARD.

North Adams, Mass.

In reply to John L. Baker on page 694 of the November issue, there is no good reason other than custom why a Corliss engine cannot run "under." Engineers have become used to the engine running "over" and it would be about as sensible to impose a change on these men as it would be to ask a right-handed man to do fancy pen work with his left hand or to ask a stenographer to operate the typewriter with her toes. When the engineer came to set the eccentric for the run-under engine he would have to imagine that the world had turned upside-down and that he was standing on his head in China. The flexibility of valve setting of a Corliss engine, however, particularly the double-eccentric engine, is so great that almost any kind of operation may be had. E. S. HAWKINS.

Chicago, Ill.

Boiler Furnaces and Combustion

In POWER for November is a letter by Peter Van Brock on the above subject, commenting on one of the same title in POWER for September, in which he seems to have overlooked the fact that in the opening sentence it is mentioned that Illinois and Iowa coal was used in these tests and the general tenor of the letter was to call attention to the value of "Dutch Oven" furnaces for such coal. The subject is of sufficient importance, I believe, to warrant further discussion and presentation of facts.

Taking up the comments in the order of their occurrence, while the statement "setting the boiler close to grates" is not as exact as he would wish, I think most engineers will understand what was meant. In this section of the country it would mean less than 24 inches from the grates to the shell of the boiler when a return tubular boiler was referred to. I have known of several boilers set 16 to 20 inches above grates and in every case when Illinois or Iowa coal was burned, poor results were obtained. In some cases the grates were lowered and a marked improvement in economy and capacity resulted. Personally I would not set return tubular boilers less than 36 inches above grates when burning these coals.

In one of the large power houses of this city are a number of water-tube boilers of the inclined-tube type, set with the front ends of the tubes seven feet above the floor line, also a number set eight feet above the floor, all served with the same make of chain-grate stoker. I am informed by the chief engineer, a man of experience and a close observer, that the eight-foot settings give better results. In another power plant having the same type of boiler set with the front end of the tubes 42 inches above the chain-grate stokers, 7 per cent. of CO_2 is the maximum obtainable; by inclosing the bottom row of tubes with firebrick tile and changing the flow of gases to that used in the Heine boiler, making practically a "Dutch Oven" furnace, 13 per cent. CO_2 is easily obtained and the capacity and economy of the boilers increased.

There are a considerable number of boilers in this city having the bottom row of tubes covered with firebrick tile and the number is increasing from year to year. From some of these boiler settings I have taken samples of gas showing as high as 17.4 per cent. CO_2.

Mr. Van Brock comments on an evaporation of 9.7 pounds of water from and at 212 per pound of combustible, assuming a probable heat value of 11,000 British thermal units for dry coal. This shows an efficiency of 64 per cent. I have obtained a higher efficiency with Illinois coal, when using an automatic stoker. In Bulletin No. 9, Engineering Experiment Station, Iowa State College, on page 7, test No. 5, will be found recorded an evaporation of 9.2 pounds of water per pound of combustible and in the Preliminary Report of the Coal-Testing Plant at St. Louis, Bulletin No. 261, page 81, test No. 55 with Iowa coal gives nearly 9 pounds; but neither of these had the "Dutch Oven" furnace. Further to show what others think of it I will quote from Bulletin No. 9 above mentioned. On page 14 Professor Bissell says "the final condition of the disengaged hydro-carbon gases as they arise from the bed of soft coal may be variable.

"If they impinge upon a relatively cold boiler shell, causing their temperature to fall below the combustion point, they will probably be swept into the flue in a half-burned condition, producing smoke and causing loss of heat.

"If on the other hand the temperature of the gases be kept up until thorough mixture with oxygen has been effected, the result is smokeless combustion and a maximum of heat developed.

"The best conditions for securing smokeless combustion with maximum steam-generating power are obtained with a proper design of coking arch, or better yet by the use of a separate furnace known as the 'Dutch Oven.'"

Professor Kent, on page 452 of his "Steam Boiler Economy," states that "In ordinary practice in the Western States an efficiency of 50 per cent. or less is not uncommon with the coal burned in ordinary furnaces. It is quite possible to raise this to 70 per cent. or even 75 per cent. with automatic stokers, furnaces surrounded by firebrick, and provision for securing the intimate admixture of very hot air with the distilled gases." The raising of the efficiency of boilers by these means from 50 per cent. to 70 per cent. would effect a saving of many millions of dollars per year, and there is more on the subject on page 635 of Kent's "Pocket Book."

That the economy of a boiler is effected by the rate of driving is too well known to require much space to quote figures. A few extracts, however, may be in order. Professor Bissell states on page 16 of Bulletin No. 9, "Table VII confirms the belief based upon other experimental data, that the most economical rate of evaporation is about three pounds of steam per square foot of water-heating surface per hour." On page 685, Kent's" Pocket Book," under table of tests of steam boilers at the Centennial Exhibition, "The comparison of the economy and capacity trials shows that an average increase in capacity of 30 per cent. was attended by a decrease in economy of 8 per cent." The subject is treated at length in Kent's "Steam Boiler Economy," Chapter IX, where, on page 230, he states, "Boilers proportioned for a rate of driving of three pounds per square foot of heating surface per hour will give about the maximum economy of all costs, including interest on investment, depreciation, etc." In the test mentioned in POWER of September the boiler developed 158 horse-power with 1600 square feet of heating surface, bringing the evaporation near three pounds per square foot per hour, hence the high efficiency.

Economy does *not* depend on the largest quantity of air you can force into the fire. Mr. Van Brock intimates this same fact where he states his friend "muzzled" the draft in the plant mentioned on Pearl street, New York.

Air contains 23 per cent. oxygen instead of one-eighth, as Mr. Van Brock says.

Steam superheated to a point of decomposition would not aid combustion of coal, as, in combining, the hydrogen would require just the amount of oxygen the steam contained. This old fallacy is treated on page 650, "Kent's Pocket Book."

The results obtained at the Pearl street plant were with anthracite coal evidently. We are not told what kind of coal was used in the marine set, but are quite certain it was not Illinois or Iowa coal.

Regarding the importance of moderate draft for best economy, there is much to be said; but as this letter is already too long, will defer until some future time.

Chicago, Ill. R. B. HOLBROOK.

Piston Packing-ring Data

During the past ten years the writer has had considerable experience in the design and application of packing rings for steam and gas engines and air compressors, and knows that the eccentric ring has quite wholly superseded the concentric ring in up-to-date shops. One of the very good reasons for this is that the eccentric ring is more uniformly flexible throughout its entire length, hence, will lie more closely to the cylinder at all points, as stated by "J. R. H." in the June, 1906, number, without being returned after it is sprung together.

The June letter says, "If there is any established rule for making eccentric rings I would like to see it published." I have compiled a table of eccentric-ring data which experience has shown to be wholly satisfactory, and I give it herewith, believing that it will be appreciated by those interested in this subject.

Franklin, Pa. E. RAY INMAN.

DIMENSIONS OF ECCENTRIC PACKING RINGS FOR PISTONS

Bore of Cylinder	Outside Diam. Open	Inside Diam. Open	Inside Diam. Closed	Width of Ring	Thickness at Points	Thickness at Back	Width of Gap
6	6 7/32	5 27/32	5 5/8	3/8	1/8	1/4	11/16
7	7 7/32	6 28/32	6 1/2	3/8	3/16	5/16	11/16
7 1/2	7 3/4	7 1/4	7	3/8	3/16	5/16	3/4
8	8 1/4	7 11/16	7 7/16	3/8	3/16	3/8	3/4
9	9 5/16	8 3/4	8 7/16	3/8	3/16	3/8	15/16
10	10 3/8	9 13/16	9 7/16	7/16	3/16	3/8	1 3/16
10 1/2	10 7/8	10 5/16	9 15/16	1/2	3/16	3/8	1 3/16
11	11 3/8	10 3/4	10 3/8	1/2	3/16	7/16	1 3/16
12	12 7/16	11 13/16	11 3/8	1/2	3/16	7/16	1 3/8
13	13 7/16	12 11/16	12 1/4	5/8	1/4	1/2	1 3/8
14	14 1/2	13 3/4	13 1/4	5/8	1/4	1/2	1 9/16
15	15 1/2	14 11/16	14 3/16	5/8	1/4	9/16	1 9/16
16	16 9/16	15 5/8	15 1/16	5/8	5/16	5/8	1 3/4
17	17 5/8	16 11/16	16 1/16	3/4	5/16	5/8	1 31/32
17 1/2	18 1/8	17 3/16	16 9/16	3/4	5/16	5/8	1 31/32
18	18 5/8	17 5/8	17	3/4	5/16	11/16	1 31/32
19	19 5/8	18 5/8	18	3/4	5/16	11/16	1 31/32
20	20 3/4	19 5/8	18 7/8	7/8	3/8	3/4	2 3/8
21	21 3/4	20 5/8	19 7/8	7/8	3/8	3/4	2 3/8
22	22 3/4	21 5/8	20 7/8	7/8	3/8	3/4	2 3/8
23	23 3/4	22 7/16	21 11/16	7/8	7/16	7/8	2 3/8
24	24 7/8	23 9/16	22 11/16	1	7/16	7/8	2 3/4
25	25 7/8	24 9/16	23 11/16	1	7/16	7/8	2 3/4
26	26 7/8	25 9/16	24 11/16	1	7/16	7/8	2 3/4
27	28	26 1/2	25 1/2	1	1/2	1	3 1/8
28	29	27 1/2	26 1/2	1	1/2	1	3 1/8
29	30	28 1/2	27 1/2	1	1/2	1	3 1/8
30	31	29 1/2	28 1/2	1	1/2	1	3 1/8
31	32 1/8	30 5/8	29 1/2	1	1/2	1	3 1/2
32	33 1/8	31 5/8	30 1/2	1	1/2	1	3 1/2
33	34 1/8	32 1/4	31 1/8	1	5/8	1 1/4	3 1/2
34	35 1/8	33 1/4	32 1/8	1	5/8	1 1/4	3 1/2
35	36 1/4	34 3/8	33 1/8	1	5/8	1 1/4	3 15/16
36	37 1/4	35 3/8	34 1/8	1	5/8	1 1/4	3 15/16
37	38 1/4	36 3/8	35 1/8	1 1/4	5/8	1 1/4	3 15/16
38	39 1/4	37	35 3/4	1 1/4	3/4	1 1/2	3 15/16
39	40 3/8	38 1/8	36 3/4	1 1/4	3/4	1 1/2	4 5/16
40	41 3/8	39 1/8	37 3/4	1 1/4	3/4	1 1/2	4 5/16

Heating Water with Live Steam

In a recent number of POWER I have just noticed an inquiry from "C. W. M." as to the heating of 1800 gallons of water per hour in a tank by means of steam at 80 pounds pressure passing through a 1-inch brass pipe coil in the tank, the temperature of the water to be raised from 140 to 180 degrees Fahrenheit. In the answer to the inquiry it is stated that the better way would be to run the water through the pipe and turn the steam into the tank. The inquirer, however, stated that the tank was used as a reservoir, so that the proposed method would scarcely fit the case. It is entirely practical to heat the water by means of a steam-pipe coil; this is one of the commonest appliances known to engineers. It is a fact, of course, that more surface is required when the steam passes through the coil than when the water does; on the other hand, "C. W. M." doubtless wants a supply of hot water from which he can draw intermittently, the average rate being 1800 gallons per hour, but the actual rate during draft being higher, and if this is true, a water coil would not meet the conditions.

To get at the pipe required I should proceed as follows: The 1800 gallons per hour are equal to 15,000 pounds per hour, and as each pound must receive 180 — 140 = 40 heat units, there will be required $40 \times 15{,}000 = 600{,}000$ heat units per hour from the steam. Steam at 80 pounds pressure gives out 885 B.t.u. per pound in condensing, and if 5 per cent. be allowed for radiation, there will be available 840 B.t.u. per pound of steam condensed. Hence, the quantity of steam required per hour will be $600{,}000 \div 840 = 714$ pounds, or 11.9 pounds per minute. This latter quantity will occupy $4.57 \times 11.9 = 54.4$ cubic feet of space. Allowing a velocity of 6000 feet per minute, a cross-sectional area of 0.0091 square foot, or 1.31 square inches, will be required in the pipe; this corresponds practically to 1¼-inch pipe.

WILLIS M. WRIGHT.

New Haven, Conn.

[Of course, the water can be heated in the way described. It can also be heated by merely turning the steam into the tank, if the latter be closed. The passage of the water through a pipe coil in a closed steam box, however, is the most efficient method, and for that reason it was recommended. Even though the tank mentioned by "C. W. M." be a reservoir, it is a simple matter to provide a closed steam box containing a pipe coil through which to draw water from the reservoir tank. Our correspondent's method of deducing the pipe diameter is rather obscure; it appears to be based on a pipe length of one foot.—EDITORS.]

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The index for the 1906 volume of POWER will be sent to any of our readers who send a postal card request with full mailing address.

Boiler "Horse-power"

It is somewhat confusing to the beginner to be told that there is no relation between the horse-power of a boiler and that of an engine, but if this information were imparted early in the course of the beginner's studies it would save him much worse confusion later on. The mechanical "horse-power" is, of course, an arbitrary unit, the equivalent of 33,000 foot-pounds of work per minute; but it serves the useful purpose of giving a criterion based on actual work per unit of time. The boiler "horse-power" is also an arbitrary unit and also serves as a criterion based on work done per unit of time; but unfortunately it bears no relation to the recognized mechanical unit of power. It is taken as the equivalent of evaporating 30 pounds of water per hour from a feed temperature of 100 degrees Fahrenheit into steam at 70 pounds gage pressure, which, when reduced to the usual basis of comparison, figures out 34½ pounds of water evaporated per hour from the atmospheric boiling point into steam at atmospheric pressure. So far as making comparisons of different boilers is concerned, this rating may be all that could be desired, but the boiler and engine are indissolubly linked together, and there is no good reason now why one kind of horse-power should be applied to the one and another kind to the other. The explanation of this practice is that the equivalent of mechanical horse-power was determined in the days of Watt and based on the work that a first-rate draft horse was supposed to be able to do; and the so called horse-power of a boiler was adopted in the early days of the higher-pressure steam engines, when the average non-condensing engine of good design required about 30 pounds of steam per hour at 70 pounds pressure for each horse-power developed in the cylinder.

The heating of 30 pounds of water from 100 degrees Fahrenheit to the boiling point and evaporating it into steam at 70 pounds gage pressure requires 33,305 British heat units, which are equivalent to 25,911,290 foot-pounds of energy, and if this work is done in one hour it is the equivalent of a trifle over 13 mechanical horse-power. In other words, for each horse-power of rating on the accepted basis, a boiler actually does over 13 horse-power. The incongruity of this is perfectly obvious, and we have pointed it out many times in the past. If there were any necessity to rate the boiler in terms of horse-power, the rating should, of course, be based on its performance as a heat transferrer, not on demands made by the engine, even if there were any such thing as an "average" engine consumption of steam.

The remedy, so long as the present absurd "standard" rating remains in force, is to avoid the use of it. For the student or beginner in steam engineering it will be found much easier to consider merely the ability of a boiler to evaporate water of 212 degrees temperature into steam at atmospheric pressure, and this is just as convenient for designing and consulting engineers, if not more so. Meanwhile, it is queer that everybody interested does not make some effort to get a rational basis of horse-power rating adopted or recognized by the leading engineering organizations, or at least get the present method of rating abolished. It has outlived its usefulness, if it ever had any.

The Maligned Commutator

Fervent, exhaustive and continuous were the curses bestowed on the commutator of the direct-current dynamo and motor during the first decade of their real commercial activity, and so inground became the habit of anathematizing this vital member of the machine that it continued in disgrace long after its makers had learned how to design dynamos and motors properly. It has been only a few years since advocates of the induction motor were putting forth as the beautiful and paramount feature of that machine the absence of that horror, the commutator; and this when commutator machines by the thousands were running day in and day out, week after week, month after month, with no trouble whatever at the brushes. Then came the poetic retribution—the development of the "compensated" alternating-current motor, which possesses characteristics that the induction motor has not and for which it has no adequate substitute, and this motor depends for its operation upon—the commutator! It is not our purpose, however, to discuss the relative merits of the induction and compensated motors, but to point out the lack of justification for the continued hue and cry against the commutator.

For at least five years there has been no engineering reason why a direct-current dynamo or motor should spark at the brushes unless brutally overloaded or persistently neglected. If any of the modern machines turned out by makers of standing fail to carry full load continuously without any sparking whatever, it is either due to obviously faulty adjustment of the brush gear or the result of an attempt by the designer to produce a low-priced machine; and no designer ever attempts this feat except under pressure from the buying public. The moral is the same here as in so many other avenues of commerce. Buy at a cheap price, and you will get a cheap article. There are manufacturers who will supply motors and

dynamos under guarantees as to sparking (and heating) which will satisfy the most exacting purchaser, but they do not sell such machines at bargain-counter prices. This class of dynamo or motor is just as reliable and just as easy to take care of as an induction motor, or any other piece of electrical machinery; a little attention is required, of course, to keep the brushes adjusted, and to renew them when worn out, but this is negligible in practice. There is no longer any operating justification for avoiding the commutator—of a decently designed machine.

The Indicator and Flue Gas Analysis

Time was, and not so very long ago, when the steam-engine indicator was a scientific instrument rather than an engineering tool; in the hands of a few exclusive experts rather than of the operative engineer and as much a mystery to the rank and file as is the polariscope or the ballistic pendulum. With its popularization the instrument has been improved in many details, has been brought within the reach of the man to whom it is most useful and numerous popular expositions of its diagrams and detailed directions for its use have placed its advantages within the reach of all and stamped the man who does not know the instrument as either a tyro or a shiftless, non-progressive member of the craft.

Just as it is now said of an engineer who is "coming along" that he has arrived at the "indicator stage" it will soon be common to remark of him who has got somewhat farther along that he has got to the "gas analysis" stage.

The Orsat apparatus is to the furnace what the indicator is to the steam engine. It tells whether or not the combustion has been perfect; if not, whether the failure to complete the process was due to a lack of air, or to a failure to obtain a sufficiently intimate mixture, or to a cooling of the gas below the point of ignition before combustion was completed, although enough or more than enough air be present. It tells the man who uses it with intelligence what has been going on in the furnace and enables him to modify operations or conditions so as to obtain improved results. We have published several times directions for making these tests, and are arranging now for a complete retelling of the story in so simple and practical a way that operating engineers may soon be talking percentages of CO_2 as intelligently as they do now mean effective pressures.

The apparatus is not so expensive as an indicator outfit, requires no more nicety of measurement than does the working up of an indicator diagram and no more brains to understand and apply its indications than are required to understand and apply the diagrams of the older instrument, and the things which it tells are of as much practical importance to the man who is running a steam plant as the knowledge as to whether the valves of his engine are set right or how many horsepower it is developing.

The Energy of Combustion

An excellent demonstration of the indirectness of the present method of generating power from fuel is furnished by the illustration of Mr. Bement's boiler-setting on page 43. The air and the gases generated from the fuel are expanded at the high temperature of the furnace, performing work by forcing back the atmosphere and are then contracted by contact with the heat-absorbing surfaces to the lowest attainable temperature, and thus to the smallest attainable volume. This is brought out strikingly in the diminishing cross-sections of the gas passages as the gas becomes cooled by successive passes among the tubes in the illustration referred to.

Suppose the air to enter the furnace at an absolute temperature of five hundred degrees (about forty-one Fahrenheit) at which temperature one pound occupies a volume of 12.6 cubic feet. Suppose it to be heated to three thousand degrees absolute (a little over twenty-five hundred Fahrenheit). The volume will be increased to 75.6 cubic feet and the energy generated (the pressure remaining equal) would be the product of that pressure by the change of volume or:

$2116 \times (75.6 - 12.6) = 133.308$ foot-pounds

Since there are about twenty pounds of flue gas per pound of coal, each pound of coal in burning does something over two million foot-pounds of work in pushing back the atmosphere.

A horse-power-hour is only 1,980,000 foot-pounds and not many plants get a horse-power-hour from one pound of coal. The fact that we have taken the full atmospheric pressure instead of the lesser pressure of the furnace does not make enough difference to detract from the strikingness of the demonstration.

In the steam boiler this work is undone, as above intimated, by cooling the gases down again, the heat passing into the water and generating steam under pressure, and thus steam is made to develop energy by working from a higher to a lower temperature level by expansion. Engines have been invented in which the combustion was carried on under pressure, the products of combustion passing to the cylinder with the steam and sharing in the expansion, but this direct application of the expansion of the products of combustion is effected more simply in the internal-combustion engine where the oil or gas is burned in the cylinder itself.

The Utilization of Waste Gases

In view of the fact that the cost of power generation in the iron and steel industry has an important influence on the net profit obtainable from the sale of the products of that industry, and, incidentally, in consideration of the relation between the price of iron and steel and the stability of other industries, it is important that close attention be given to the matter of increasing the economy of power production in connection with steel and iron works. Since it has become recognized in a general way that the gases which formerly were allowed to escape from blast furnaces and coke ovens can be turned to account in the way of power generation, it becomes of practical interest to study the possibilities in this direction and the comparative merits of different methods of thus applying the surplus gases. The article in another part of this issue by Mr. Junge deals very lucidly and thoroughly with this subject, and brings out the salient factors which affect the generation, transmission and distribution of power derived from the waste gases of blast furnaces and coke ovens and those obtained from gas producers supplied with fuel from culm banks or other waste.

Lap-welded Boilers.

Massachusetts, the only State in the East which requires boilers to be inspected and engineers to be examined and licensed, has been visited by another disastrous boiler explosion, that at West Lynn. Particulars and illustrations will appear in our February issue. It is another of those lap-seam failures which are responsible for the Brockton, Fall River and New Bedford explosions and which present a problem alike for the steam user, the boiler insurance company and the legislator. Sixty per cent. of the boilers in use in the East are of this type. Their displacement by those having the safer butt-strapped joint would involve an immense financial sacrifice, but the alternative is a continuation of these explosions with a sacrifice of human life, an interruption to industry and a property loss which in the aggregate will far exceed the sacrifice involved by their rejection. At least the further manufacture of the shell boiler with lap-riveted joints should be prohibited in diameters of over forty-two or at most forty-eight inches and for pressures of over one hundred pounds; if they are manufactured the insurance companies should refuse to carry them, and whatever is done or is not done the man who knowingly and willingly permits such a boiler to run under pressure when he is in a position to prevent it is morally responsible for the murders it commits when it goes off.

NEW THINGS FOR THE POWER PLANT

H. & M. Combination Hand and Automatic Stop Valves

The H. & M. Valve Company, 25 Broad street, New York, is building a line of globe and angle valves of the combined manual and automatic type, the features of which are illustrated herewith. Fig. 1 shows the outward appearance of the angle valve and Figs. 2 and 3 give vertical sections of angle and globe valves, the former of the universal type and the latter of the gravity type for vertical mounting only. The object of the automatic feature, of course, is to close the valve in the event of pipe breakage or any other accident

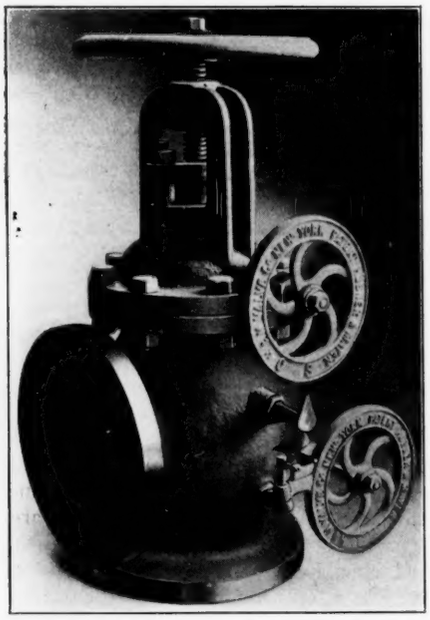


FIG. 1. H. & M. ANGLE VALVE

which would allow the steam to escape freely to the atmosphere.

Referring to Fig. 2, the vertical spindle *A* carries loosely on its lower end the main valve disk *B*, which is provided with a long hollow shank bored out to fit the spindle *A*. A lever, forked at its inside extremity, spans a recess or annular groove on the outside of the hollow shank. This lever is keyed to a short horizontal spindle which is hidden in Fig. 2 by the spindle *A*; the horizontal spindle passes through a stuffing-box in the side of the valve body and carries on its outer end the upper of the two side hand-wheels shown in Fig. 1. By means of this hand-wheel the valve disk *B* may be raised or lowered when the spindle *A* is backed out, which is its normal position when the valve is in service. Ordinarily, the valve disk *B* is lifted by the steam pressure from below, the flow of steam being from the bottom opening to the side opening; should there be a break between the boiler and the valve, the disk *B* will drop to its seat and prevent any back flow of steam from the main.

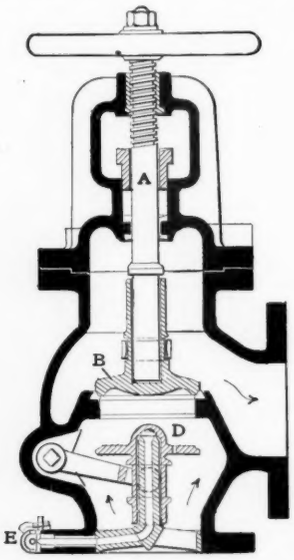


FIG. 2. H. & M. UNIVERSAL ANGLE VALVE

A second valve disk *D* is mounted on a vertical stem through the center of which is a passage to the atmosphere; this passage is closed by the disk when it is in the position shown. Should there be a break in the system beyond the valve, causing an abnormal rush of steam through it from

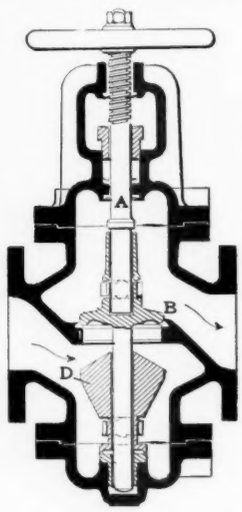


FIG. 3. H. & M. GRAVITY GLOBE VALVE

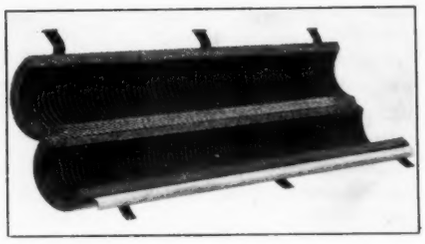
the boiler, the increased velocity of the steam will lift the disk *D* to the under side of the central seat, closing the main passage through the valve and at the same time opening the small passage in the guide-stem of the disk *D*; steam escapes to the atmosphere through this small passage *E*, giving notice that the lower disk has closed. Once this has occurred, the valve cannot be opened again without screwing down the vertical spindle *A* and seating the disk *B*. There is a tiny orifice drilled through the disk *D*, and this equalizes the pressure on opposite sides of the disk when the disk *B* is closed, allowing *D* to return to its seat, which it does under the influence of the preponderance of steam pressure on its upper surface, irrespective of the effect of gravity. It is this feature which permits the valve to be mounted in any position. The outlet *E* is adjustable so that the conditions under which the disk *D* will close may be predetermined. The disk *D* may be raised or lowered from the outside by means of the lower hand-wheel shown in Fig. 1 and the lever shown in Fig. 2.

Fig. 3 shows the type of valve for mounting in a vertical position only, a globe valve being shown in this case. The only difference between this type and the universal type just described is that the valve *D* is sufficiently heavy to seat itself by gravity under normal conditions; it is drawn to its seat by an abnormal rush of steam, just as in the previous case. The disk *B* is exactly the same in both types. The hand-wheels controlling the disks *B* and *D* are for use in testing the valve only; the manipulation otherwise required is limited to the screwing down of the spindle *A* after an accident which has caused the disk *D* to seat, and unscrewing the spindle after the cause has been removed.

The spindle *A* is provided with a collar immediately above the shank of the disk *B* which closes off the inner end of the stuffing-box steam-tight when the spindle is unscrewed all the way, thereby permitting the replacement of the main packing in the stuffing-box while the valve is in service.

Asbestocel Pipe Covering

The accompanying engraving illustrates a brand of pipe covering for hot-water, steam and hot-air heating systems known



ASBESTOCEL PIPE COVERING

as the Asbestocel covering, and manufactured by the H. W. Johns-Manville Company, New York. This covering is said to be absolutely fireproof, and to meet the fire underwriters' requirements.

It is made entirely of asbestos, and is

said to be an improvement over ordinary air-cell coverings, in that the corrugations run around the pipe (as indicated in the illustration) instead of lengthwise, thus preventing all circulation of air and consequent radiation of heat. This covering is also claimed to be unusually strong and durable, owing to the fact that it is built on the arch form of construction.

Asbestocel covering is made in sectional form for pipes, in sheets and blocks for large heating surfaces and paper for hot-air pipes.

"Cookson" Horizontal Steam or Oil Separators

In the separator shown in Fig. 1 the entering steam is divided, an equal amount passing on each side of the star baffle plate. In its course through the separator,

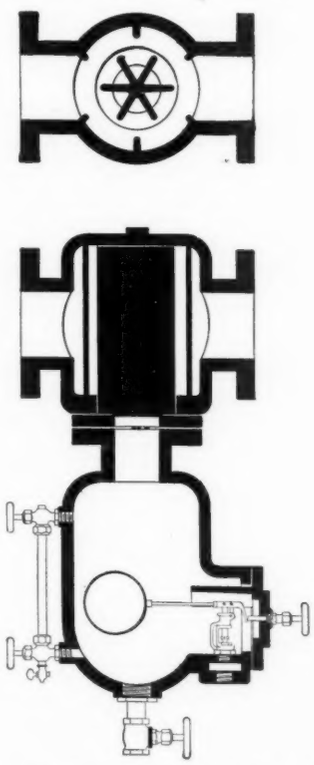


FIG. 1. THE COOKSON SEPARATOR WITH TRAP ATTACHED

the steam is given a zig-zag movement, first engaging the baffle and then the ribs on the shell, the water or oil passing down the bottom of the head when it drains through openings into the bowl or well at the bottom. This water cannot again be picked up by the steam current.

One especial feature in the construction of this separator, and one that will be appreciated by engineers, is the fact that the separator can be taken apart without breaking the steam piping. By removing the bolts, the bowl comes off; the star baffle then is removed. The entire inside of the head of separator is now accessible. Oil separators especially must at times be cleaned and any simplification of the process is an advantage. The receiver is made with a float trap when desired, so that it discharges automatically.

It is made by the D. T. Williams Valve Company, of Cincinnati, Ohio, who also

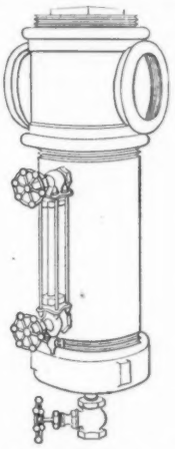


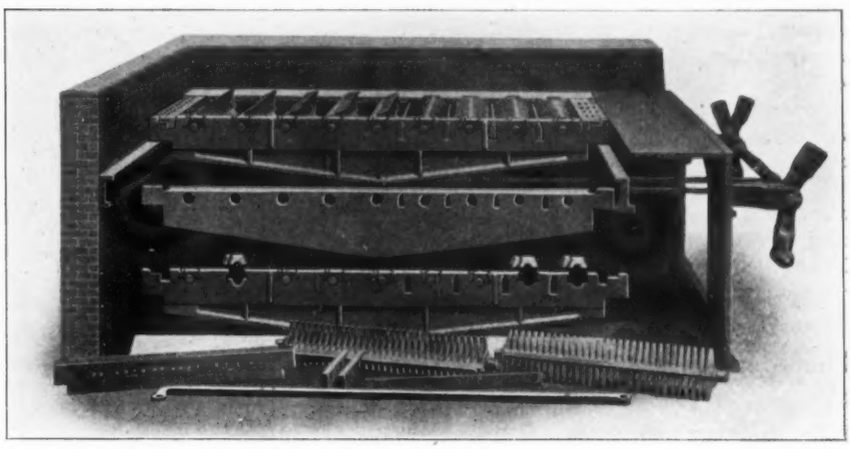
FIG. 2. THE COOKSON PIPE SEPARATOR

make a vertical separator which is particularly adapted for use as a combined separator and receiver. Another of their products is the pipe separator shown in Fig. 2. This is a combination of pipe fittings, a specially designed plug screwed in at the top forming the baffle.

These plugs are sold separately if one desires to make up the rest of the separator himself.

The Improved Neemes Grate

This grate, designed for use in furnaces six feet in length and over, is a combination of the well-known shear-cutting grate of the builders for the forward section, with a shaking and dumping grate for the back



IMPROVED NEEMES GRATE

section. The front or thin part of the fire can be kept clean without loss of combustible and with the least possible disturbance, while the thicker fire at the back can be clinkered from the bottom, or the fire can be raked forward and the clinkers pushed onto the back section and dumped. The utility of the combination will appeal to the man who has charge of furnaces. The bars are locked in place with the double-wedge lock box described in detail in our issue of August, 1905. With this box the shakers are held in place, cannot rise into the fire, and yet any member of the grate is as easily removable as though it only lay in the bearing. The grate is adapted to the smaller sizes of anthracite as well as soft coals, and enables them to be used without a slice bar in the rear of the furnace. It is made by Neemes Brothers, Troy, N. Y.

Water Circulators for Steam Boilers

The accompanying engravings illustrate a line of apparatus designed to increase the circulation of water in shell boilers and manufactured by H. Bloomsberg & Co., 425 North Carey street, Baltimore, Md. Fig. 1 represents a circulator which

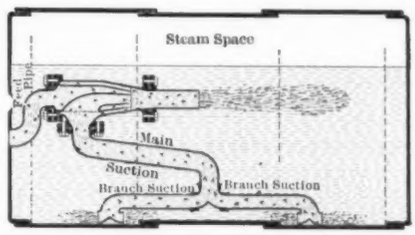


FIG. 1. FEED-WATER CIRCULATOR

depends for its operation on the force of the incoming feed-water. This is delivered into the boiler through a nozzle located near the water level and composed of three concentric tubes; the feed-water passes through the intermediate one of these and induces a flow of water through the central and outside tubes, which take their supply from the bottom of the boiler shell through the pipes designated "Main Suction" and "Branch Suction." This, of course, takes the colder water from the bottom, warms it to some extent and delivers it at the top along with the entering feed-water.

The apparatus shown in Fig. 2 is intended chiefly for warming up the water of a boiler while it is being fired up preparatory to putting it into commission. A jet of steam is admitted from some other boiler to a compound nozzle at the junc-

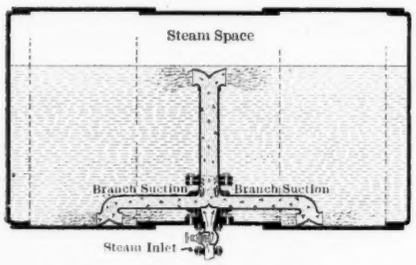


FIG. 2. STEAM-JET CIRCULATOR

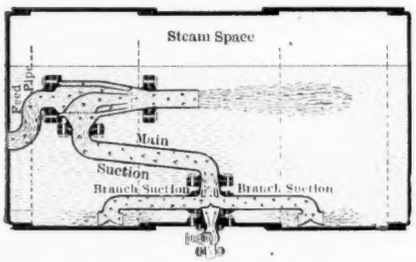


FIG. 3. FEED-WATER AND STEAM-JET CIRCULATOR

tion of the two branch suction pipes and this causes a flow of water from the bottom to the top of the boiler, heating the water as it passes upward. Thus, the water is both circulated and heated at the outset instead of having to wait for the fire to heat it and produce natural circulation—which, incidentally, is not always as thorough as might be desired.

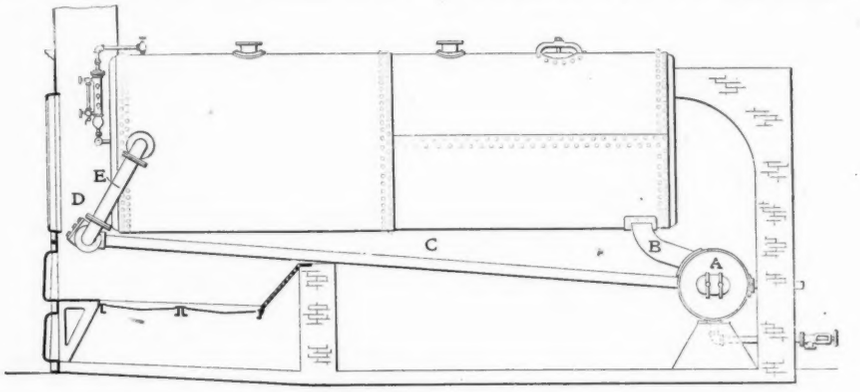
Fig. 3 shows a combination of the two pieces of apparatus just described; to the feed-water circulator of Fig. 1 is added the steam jet of Fig. 2. While the heating circulator is intended to be used ordinarily before the feed-water circulator comes into commission, there is no conflict if both are used simultaneously, since the movement of water induced by both is the same.

In all three engravings the piping is drawn to a much larger scale than the boiler shell, in order to show the details of construction without making the engravings unnecessarily large.

The Montauk Steam Generator

The "generator" as distinct from the boiler itself consists of additional heating surface provided by pipes disposed in the furnace and combustion chamber, and so connected to the boiler that there shall be a continuous circulation of water through them. The files of POWER contain descriptions of a number of forms of the device, which may be applied to existing boilers and offers an expedient for increasing capacity when it is impracticable to put in additional boilers.

The latest form of the generator is that offered by the Montauk Engineering Company, 50 South street, New York. The drum *A* is of sufficient size to permit easy entrance for cleaning or other purposes, and is attached to the rear bottom of the boiler by the steel casting *B*, all the joints at this point being riveted. The tubes *C* are sufficiently inclined to insure rapid circulation and are expanded at the front end into the steel header *D*, extending across the entire width of the



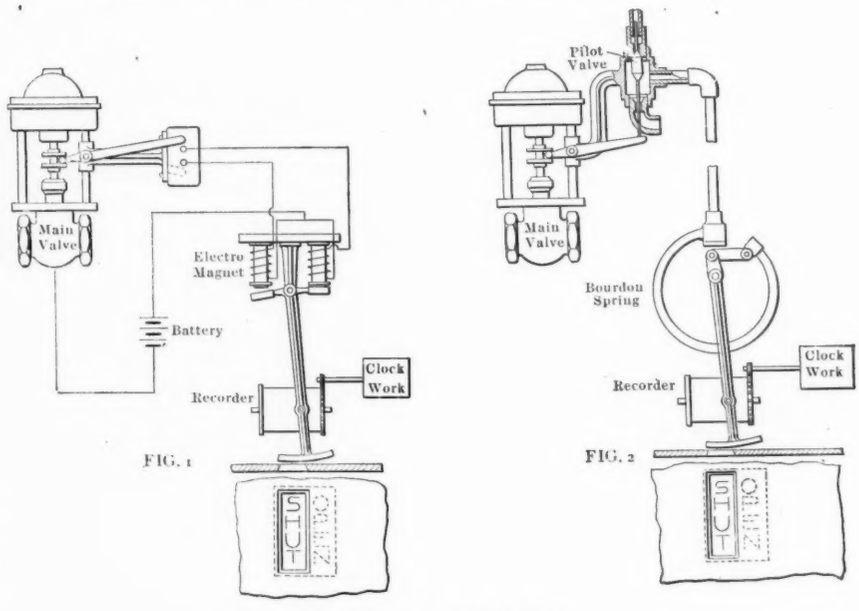
THE MONTAUK STEAM GENERATOR

furnace and capable of being opened at both ends for cleaning purposes. Risers *E* connect the ends of this header with the boiler just below the water line. This arrangement permits free movement of the tubes and relieves the whole structure from possible expansion strains.

The Gould Valve Indicator

This indicator, which may be located at any point, however remote from the valve itself, is designed to show at a glance whether the valve to which it is connected is opened or closed. The chief engineer or superintendent may, for instance, have upon the wall of his office such indicators connected with the principal valves of the plant and be able to tell at a glance the conditions under which the plant is working. Since any number of indicators may be operated simultaneously from one valve, indicators may be located at any number of points for the same valve. When desired, a recording apparatus may be attached which will tell in what position the valve was at any time during the day.

The indicator may be operated electrically, as shown in Fig. 1, or mechanically, by means of the pressure controlled by the valve itself, as shown in Fig. 2. A lever attached to and operated by the valve-stem makes the contact necessary to move the indicator by means of the electro-magnet in the first instance, or operates the pilot valve which admits pressure to the Bourdon spring by which the indicator is operated in the second case. The operation is obvious from the drawings. F. E. Gould, manager of the Williams Gauge Company, Fisher Building, Chicago, is the inventor.



THE GOULD VALVE INDICATOR

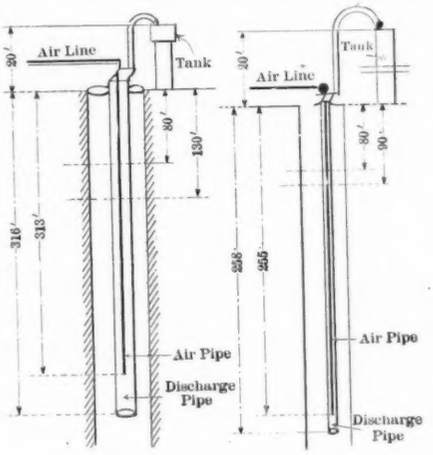
The value of rubber and gutta percha imported into the United States has grown from less than $16,000,000 in 1896 to more than $50,000,000 in 1906.

INQUIRIES

Questions Are Not Answered Unless They Are of General Interest and Are Accompanied by the Name and Address of the Inquirer

Air-lift for Brine

We have at our plant a brine well 1000 feet deep. The casing for the first 300 feet is 5 inches and 4 inches the rest of the way down. The brine tests 100 and stands up to within 80 feet of the top of the well. I wish to install an air-lift and want to know whether it could be done successfully or not. I want at least 60 gallons per minute delivered against a 20-foot head. What size discharge and air pipe should I use and how deep should the pipes be submerged? I have a 20-ton vertical ice machine here and intend to



AIR-LIFT FOR BRINE

use it to furnish pressure to test with. Would 2-inch pipe with ½- or ¾-inch pipe for air inside do? What are the proper proportions of depth of well, depth of submersion and head in full for an air-lift?

R. H. R.

The data as far as submitted are favorable to the successful operation of the air-lift system, provided, only, that the well yields the required quantity of brine without falling off beyond the economical limit of the lift. The air-lift will handle the full yield of any well, but if the well only produces 50 gallons per minute, the capacity of the pump is limited to that amount.

The only factor affecting our calculations is this same question of the pumping level, or the drop that will occur when the well is being pumped at the rate of 60 gallons per minute.

Let us assume that this lowering of the static level will amount to 50 feet, then

Lift above surface	20 feet.
Normal brine level in well	80 "
Drop when pumping	50 "
Total head	150 "

The best ratio of submergence to lift, under such conditions, is 55 per cent., meaning that 55 per cent. of the total length of the air-lift piping must be submerged, which is equivalent to 316 feet of discharge pipe and 313 feet of air pipe in the well, the end of the air pipe or "foot piece" being about 3 feet above the end of the discharge pipe.

The central air-pipe system should be used, discharge pipe 3 inches diameter, air pipe 1 inch diameter.

This work will call for 54 cubic feet of free air per minute at 95 pounds working pressure, equaling 9.3 indicated horse-power, in the steam cylinder of the compressor. We must at first have sufficient air pressure to start the water, after which the level recedes until the pumping level is reached.

The starting pressure in this case will be about 122 pounds, the working pressure as above stated 95 pounds, these pressures being based on lifting brine of full saturation.

Taking another view of the case, let us assume that the well will yield 60 gallons per minute without falling off more than 10 feet below the normal, making:

Lift above surface	20 feet.
Normal brine level	80 "
Drop when pumping	10 "
Total lift	110 "

I would advise here 60 per cent. submergence, which would demand 258 feet of discharge and 255 feet of air pipe in well.

The central air-pipe system should be used in this case also and the same diameters of discharge and air pipe.

The power requirements are 37 cubic feet free air per minute, starting pressure 91 pounds, working pressure 86 pounds, representing 6.1 indicated horse-power in the air-compressor steam cylinder.

Low-voltage Lamps on High-voltage Circuits

If five 100-volt lamps are connected on a 500-volt line they will burn as long as all of them are sound; if one gives out it will put out the whole five. Is there any way to avoid this? R. R.

No practical way. It is possible to put in automatic devices which will insert a dead resistance in place of any lamp that fails, but this is prohibitively expensive and not wholly reliable.

Comparative Evaporation per Pound of Coal.

Please tell me how to compute the difference in evaporation per pound of coal as shown by the following tests:

NO. 1.

Duration of test	5 hours.
Coal burned, total	600 lbs.
Water evaporated	4200 lbs.
Steam pressure, gage	80 lbs.
Feed-water temperature	150 deg.

NO. 2.

Duration of test	5 hours.
Coal burned, total	925 lbs.
Water evaporated	8075 lbs.
Steam pressure, gage	85 lbs.
Feed-water temperature	156 deg.

Reduce the evaporation to heat units, thus:

TEST NO. 1.

Heat per pound of steam	1180.44 B.t.u.
Heat per pound of feed-water	118.3 B.t.u.
Heat received from furnace, per pound of steam	1062.14 B.t.u.
Water evaporated	4200 lbs.
Total heat rec'd from furnace, 4200 × 1062.14 = 4,460,988 B.t.u.	
Coal burned	600 lbs.
Heat in steam per pound of coal burned, 4,460,988 ÷ 600 = 7434.98 B.t.u.	

Test No. 2, treated the same way, shows 9229.42 B.t.u. per pound of coal burned, or a gain of 1794.44 B.t.u. per pound of coal. To reduce B.t.u. to pounds of water evaporated "from and at 212," you must divide the B.t.u. by 965.7, the latent heat in steam at that temperature. In the tests cited, the evaporation from and at 212 is 7.685 pounds of water in the first test and 9.539 pounds in the second test, per pound of coal. These tests are incomplete, however, because you have not taken into consideration the quality of the steam and the moisture in the coal.

A Short-circuit

Why is a solid metal connection between the terminals of a transformer not a short-circuit? C. S.

It is, if its resistance is relatively small, and if it is not in the form of a coil. If it is a coil, its inductance will prevent it from actually short-circuiting the transformer. Any connection is a short-circuit if its resistance or impedance is less than equal to $E \div I$, E being the volts at the terminal of the connection and I the maximum current allowable in the other part of the circuit.

Weight of a Cast-iron Ball

What diameter would a cast-iron ball need to have in order to weigh 600 pounds? P. N. S.

Cast-iron weighs 0.2604 pound per cubic inch; 600 pounds will measure, therefore, $600 \div 0.2604 = 2304$ cubic inches. The relation between the diameter and the volume of a sphere is

$$\sqrt[3]{\frac{Vol.}{0.5236}} = Diam.$$

Applying this to your case gives 16.4 inches as the required diameter.

Book Reviews

"Elements of Gas Engine Design." By Sanford A. Moss. Published by D. Van Nostrand Company, New York. Linen-covered boards; 197 pages, 3½x6 inches; illustrated. Price, 50 cents.

The ratio of material to space in this little book is enormous; the author covers practically the whole ground indicated by the title in a space equivalent to about twenty-six pages of POWER. One or two expressions of opinion are open to question, such as "The cost of a producer-gas plant is about the same as that of a boiler of equal power," but the technical discussion is admirable throughout. Other American builders of producers will doubtless fail to be pleased by the assertion that the Taylor gas producer is the only one "used to any extent in this country for gas engines," but the obvious inaccuracy in this statement does not affect the value of the book to anyone interested in gas-engine principles. The statement on page 109 to the effect that it makes no difference whether or not the constituents of the explosive mixture are thoroughly mixed prior to their admission to the cylinder is the only one of an engineering nature that the reviewer finds questionable. The short discussion of the theories of heat suppression is particularly good. The scope of the book extends from the elementary chemistry of fuels and combustion to the practical proportioning of engines, including, of course, their principles of operation.

"Combustion and Smokeless Furnaces." By Joseph W. Hays. Hill Publishing Company, New York. 104 pages, 6x9 inches. Illustrated. Cloth. Price, $1.50.

Mr. Hays is a graduate of Cornell and a consulting engineer who confines his practice to cases involving fuel and combustion. His book is designed to put the layman in possession of information already familiar to the engineer, in order that he may deal intelligently with the smoke abatement problem, often rendered pressing by the passage of drastic ordinances. In the first chapter upon "Heat and Combustion" the reader is given an adequate idea of heat, its measurement and its mechanical equivalent. "Combustion and the Boiler Furnace" reviews simply the chemical reactions which take place in the furnace, the amount of heat produced and the resulting temperature, the effect of air supply and furnace efficiency. Chapter III treats of the steam boiler in its relation to the furnace, considers the dampening effect on the relatively cool surfaces upon combustion, and suggests enveloping tile and Dutch ovens. "The Chimney Evil" is considered from the hygienic and the economical points of view. The requisites of a good smokeless furnace are stated and mechanical stokers discussed. A chapter is devoted to "Hand-Fired Furnaces," and the final chapter presents the conclusions of the author as to the conditions necessary to complete combustion, the further conditions necessary to efficiency therewith, various sources of damage and some suggestions as to the value of tests. The matter is presented in a popular way with no mathematics or chemical symbolism beyond the comprehension of the average reader and appears to effectively fulfil the purpose for which it was written.

Corliss Engine Builders' Association

The leading builders of Corliss engines of the United States met at Philadelphia recently and formed an association of which J. C. Hooven, of the Hooven-Owens-Rentschler Company, is president; Gardiner C. Sims, of the William A. Harris Steam Engine Company, vice-president, and Randolph T. Ode, of the Providence Engineering Works, secretary. The association will deal with matters general to the interests of its organizers, including standardization of parts, etc.

In the recently issued number of the *Valve World* R. T. Crane says some very pertinent and pointed things in connection with Mr. Hill's recent plea for reciprocity with Canada, regarding lost opportunities for trade with that country and the general incongruity of a tariff wall to protect overgrown industries.

Personal

W. L. Goddard, for several years chief engineer at the McCormick twine mills of the International Harvester Company, Chicago, was recently promoted to take charge of the power equipment of the company's newly acquired plant at Akron, O. Mr. Goddard was an organizer of Robert Fulton Association No. 28, N. A. S. E., and prior to going to Akron members of that association presented him a substantial token of their esteem.

Obituary

Elwood E. Taylor, the inventor of the Taylor stoker described in our December issue, died at Roxbury, Mass., on November 15, after an operation for appendicitis. Mr. Taylor was but thirty-five years of age, most of his business life having been spent in the automatic furnace business in connection with the Jones and American Under-Feed stokers and more lately with the American Ship Windlass Company, which is developing the stoker of his own invention.

Business Items

The Morris Machine Works, Baldwinville, N. Y., are sending out a calendar for 1907.

The Harrisburg Foundry and Machine Works advise us that there are now more than 1,500,000 horse-power of Fleming engines in successful operation.

The Nagle Engine and Boiler Works, Erie, Penn., have undertaken the manufacture of high-grade Corliss engines, of which they are now prepared to build a complete line.

T. C. Maxfield has been appointed manager of the New York office of the Weber Steel-Concrete Chimney Company, at 507 Singer building, Broadway and Liberty street.

The B. F. Sturtevant Company, Boston, Mass., report recent sales of 11 generating sets, 4 inclosed forced lubrication engines, three economizers and nine mechanical draft installations.

The steam engine department of the Allis-Chalmers Company report recent sales of 37 Reynolds-Corliss engines of various types—also a number of orders for condensers, hoisting, blowing engines, etc.

The D. T. Williams Valve Company, 904 Broadway, Cincinnati, Ohio, have got out a large wall calendar for 1907, with one week on each sheet. They will send one of these calendars to any of our readers who send their name and address.

The Snider-Hughes Company, of Cleveland, and the Canton Pump Company, of Canton, Ohio, have consolidated under the name of The Canton-Hughes Pump Company. The Snider-Hughes plant has been removed to Canton where the factory and general offices of the new company will be located.

The Buckeye Boiler Skimmer Company, Toledo, Ohio, received on October 27, a letter from the superintendent of the Lucas county Infirmary, Toledo, Ohio, in which he says that "the three Buckeye automatic skimmers we put in our boilers has given us entire satisfaction, in fact far beyond our expectation."

The Ohio Injector Company, Wadsworth, Ohio, has issued a memorandum book for 1907. It contains some useful information for engineers, maps of the United States, European countries, etc., and blank pages for memoranda. The book is bound in leather and is sent upon receipt of 25c. to cover costs, etc.

The Pittsburgh Feed Water Heater Company, Frick building, Pittsburg, Pa., has just received its new catalogs for 1907, and a copy of same will be sent to anyone on application. The new catalog is much larger than the last issue, containing additional cuts, testimonials and list of users of Pittsburgh heaters.

The Harbison-Walker Refractories Company, of Pittsburg, Pa., has taken over the selling organization of the Portsmouth Harbison-Walker Company and in the future all brands of brick, etc., made by the Portsmouth Harbison-Walker Company will be sold by the Harbison-Walker Refractories Company direct from its Pittsburg offices.

E. H. Kellogg & Co., 243 South street, New York, N. Y., have received a letter from the Nordberg Manufacturing Company, Milwaukee, Wis., in which they say that they find the Kellogg cylinder oil to be the cheapest oil they know of, despite its greater first cost. They say it has answered every purpose when all other oils have failed.

James Bonar & Co., Inc., of Pittsburg, have on hand at the present time 13 complete automatic continuous oiling systems to install. They have just closed with the Carnegie Steel Company for three systems for the Duquesne Steel Works, one for the Jones & Laughlin Steel Company, and one for the Pittsburg & Butler Street Railway Company, at Renfrew, Penn. Their catalog on oil filtration will be sent to anyone interested.

The Atlas Engine Works, of Indianapolis,

Ind., have just closed an extraordinary year, claiming to have increased their sales 85 per cent. and shipments 90 per cent., and it is their proud claim that they have so taken care of the remarkable increase in business as not to have had at the end of their fiscal year one single past due order on their books. They say that during the ensuing year their aim will be to give quick shipment of goods wanted.

P. A. Young has opened an office in the Boyce building, Charleston, W. Va., for the sale of mine, mill and electrical supplies. Among some of the concerns Mr. Young represents are the Platt Iron Works, Dayton, O., Penberthy Injector Company, Detroit, Mich., H. W. Johns-Manville Company, New York, and the Rockwood Manufacturing Company, Indianapolis, Ind. Mr. Young would be glad to hear from other manufacturers desiring representation in that city.

The Curry Oil and Supply Company has taken larger quarters at 284 Franklin street, Boston, its former store being at 294 in the same block. The company consists of C. W. Curry and his three sons, Sidney L., Malcolm Leslie, and Charles H. The Currys are born to the oil business, being the descendants of the head of the successful Boston house of Daniel Curry & Company, and the present business is practically an extension of that of the older house to include a full line of packing and engineering supplies.

Wm. B. Scaife & Sons Co., Pittsburg, Penn., manufacturers of "We-Fu-Go" and "Scaife" water softening and purifying systems, have issued an eight-page folder, the cover pages being reproductions of a scaled boiler-tube and of one which is clean. On the scaled tube is lettered "Is this Economy?"—on the clean tube, "This is Economy." The inside of the folder contains statements from some of their users regarding the savings effected in fuel and in the operation of their boilers. The folder also contains illustrations of some of their many installations.

In order to handle to the best advantage their large and rapidly increasing business in that section, the H. W. Johns-Manville Company, New York, have opened a New Orleans branch. This will consist of a large retail store, offices and warerooms, located at the corner of Baronne and Perdido streets. The Boston branch of the H. W. Johns-Manville Company have moved into a new building at Nos. 55-57-59 High street, Boston, Mass. This entire building, which comprises seven floors, will be occupied by the offices, sales and shipping rooms of the company.

The American Spiral Pipe Works, manufacturers of Taylor's spiral riveted steel pipe, have recently completed a new plant at Chicago, Ill. The plant covers twenty acres, the property comprising four city blocks between Forty-sixth and Forty-eighth avenues, south of Fourteenth street. The power plant contains two Munoz boilers, supplying steam to a Filer & Stowell Corliss engine direct connected to a National generator. Other apparatus in the power plant is a Stilwell open heater, Gardner feed pumps and a Buffalo fire pump. All piping and valves were furnished by the Pittsburg Valve Foundry and Construction Company.

The Buffalo Forge Company, Buffalo, N. Y., have closed recent contracts for the installation of four mechanical draft equipments. This company has received from the Carnegie Steel Company an order for ten electrically driven gas-cleansing fans, each of a capacity of 15,000 cubic feet of gas per minute, and from the Illinois Steel Company, an order for two gas-cleansing fans. This company has just issued a booklet on disk wheels, showing cuts of types particularly suited for cooling tower application, and they give tables of capacities in cubic feet of air per minute at speeds from 100 to 700 revolutions per minute. The wheel diameters vary from 18 to 84 inches in diameter.

The Robb-Mumford Boiler Company, South Framingham, Mass., have been awarded the contract for furnishing twenty-four 84x19′ 6″ horizontal return tubular boilers to be installed at Culebra, Isthmus of Panama, in connection with work which the Isthmian Canal Commission is doing at that point. They have also secured a contract from the Boston & Albany Railroad to build six new locomotive boilers for some of their largest freight engines. They are also building three 200-horse-power boilers of their special internally fired type for the New Orleans Ice Company, New Orleans, La., and have recently installed a 150-horse-power boiler of their internally fired type at the plant of the Ames Shovel and Tool Company, North Easton, Mass., and a 125-horse-power boiler of the same type at the plant of the Camden Woolen Company, Camden, Me. The last mentioned installation being the second boiler of the same type at the Camden Woolen Company's plant.

A fire occurred in the early morning hours of November 5, at the Corliss Steam Engine Works, of the American and British Manufacturing Company. It has occasioned wild and exaggerated rumors and comment, which were entirely unfounded. The fire was confined to the Diesel engine testing building, which was entirely destroyed. The rumors arose from the supposition that the oil used in the engines was the occasion of the fire and also because there was an explosion of some violence. The facts are that the building appears to have ignited from the outside and the small amount of oil used in the Diesel engine testing had no appreciable effect in adding to the fire. The explosion was due to compressed air in flasks which, under the increased pressure of the heated air and the softening of the metal of the flasks, naturally exploded. The effects of the explosion were local and very limited, as is the well known effect of compressed-air explosions. There will be no material interference with the output of the works in any department.

New Equipment

The city council of Gridley, Cal., is considering the construction of an electric-light plant.

The Manufactures' Light and Power Company, of Fort Wayne, Ind., has been incorporated. Capital, $50,000. Directors: S. S. Hanna, John E. Roth, etc.

The Central Colorado Power Company has been incorporated to develop water power on Grand river to furnish electricity to Leadville, Cripple Creek and other mining towns. Myron T. Herrick, of Cleveland, Ohio, is president.

A combination power, light and heating plant is being erected in the rear of the Brown-Lipe Gear Company's factory at Syracuse, N. Y., to be used jointly by the Gear Company and the Kemp-Burpee Manufacturing Company.

The mechanical division of the Department of Water Supply, Gas and Electricity, room 932, No. 13 Park Row, New York City, will be glad to receive catalogs of pumps, boiler-room fittings and supplies, including tools, hardware, pipe and fittings, rubber goods, packing, paint, etc.

The Chelan Electric Company, of which J. T. McChesney, of Everett, Wash., is president, will erect a power plant for the operation of an electric railroad. The power will also be utilized in supplying the town of Chelan with electric lights and water and in lighting several other towns.

Help Wanted

Advertisements under this head are inserted for 50 cents per line. About six words make a line.

WANTED—Men capable of selling high-grade steam specialty. Address M. M. Co., Box 189, POWER.

Situations Wanted

Advertisements under this head are inserted for 25 cents per line. About six words make a line.

MECHANICAL ENGINEER, 15 years' experience erecting, operating, consulting, contracting, understands Spanish, wishes engagement with Spanish-American concern, superintending erection or operation. Address Box 190, POWER.

I DESIRE TO REPRESENT, with office in St. Paul, a reliable firm, manufacturing, power-plant equipment or supplies. Can satisfy as to ability and character. Address Box 187, POWER.

CHIEF ENGINEER of large steam electric plant wishes to change; practical and technical education; expert with indicator and valve gears; excellent references; all letters answered. Box 178, POWER.

GENERAL FOREMAN or assistant superintendent of shop producing valves, fittings and piping for high-pressure power plants; 12 years' shop experience as toolmaker, foreman and designer; excellent references. Box 188, POWER.

Miscellaneous

Advertisements under this head are inserted for 50 cents per line. About six words make a line.

GREENFIELD VERTICAL ENGINES lead the market. Send for circular. Address Greenfield Engine Works, Harrison, N. J.

WANTED—One 14x28x42 or 36 about 250 to 300 h.p. cross-compound Corliss engine in good repair. Tennant & Hoyt, Lake City, Minnesota.

ENGINES AND BOILERS, ⅛ to 2-h.p. engine castings in sets. Model and general machine work. Sipp Electric and Machine Co., Paterson, N. J. Catalog 4c.

PATENTS—H. W. T. Jenner, patent attorney and mechanical expert, 608 F St., Washington, D. C. I make free examination and report if patent can be had, and exact cost.

PATENTS SECURED PROMPTLY—Highest references from prominent manufacturers. Write for Inventors' Hand Book. Shepherd & Parker, 512 Dietz Building, Washington, D.C.

IF INTERESTED in up-to-date water-tube boilers, investigate the merits of those manufactured by the East End Boiler Works, Detroit, Mich. See their advertisement on page 158.

DO YOU WANT machinery constructed? If so, get our estimate. Designing, pattern-making and constructing in all its branches. Address Greenfield Steam Engine Works, Harrison, N. J.

The annual meeting of the stockholders of the Hill Publishing Company, for the election of five directors for the ensuing year, and for the transaction of such other business as may properly come before the meeting, will be held at the office of the company in the Hallenbeck Building, 497-505 Pearl street, Borough of Manhattan, New York City, N. Y., on Monday, January 14, 1907, at 12 o'clock noon. Dated, New York City, Dec. 17, 1906. F. R. Low, Secretary.

WE HAVE on hand a few bound volumes of POWER for the years 1902, 1903, 1904. We need the space they occupy, you need the books if you are interested in power plant engineering. In order to move them quickly we will send them to the men who send us $2 a volume; state the year you want. The purchasers pay express charges. When these volumes we have on hand are gone there will be no more. Money received too late will be returned the same day. POWER, 505 Pearl street, New York City.

LOOKING FOR CAPITALISTS to invest to form a stock company to put these machines on the market. Having just finished a small model of a machine where I can demonstrate from the machine and drawings that the machine will produce more power than what is put into it, and also can demonstrate from a machine which I have just finished that I can transfer a horseshoe-magnet force into a revolution and get more power out of it than what is put into it. E. C. Reiter, 5 Stone street, Rockville, Conn.

For Sale

Advertisements under this head are inserted for 50 cents per line. About six words make a line.

FOR SALE — A slightly used engineering library, bound ¾ leather, 10 vols.; cost $50. Will sell for $15. Address B. R. T., Box 425, Chicago.

FOR SALE—Cyclopedia of Engineering, 4 volumes, bound in half morocco, slightly used; cost $19. Will sell for $5. Address T. M., Box 425, Chicago.

FOR SALE—Foundry and machine shop with equipment (electric) on street-car line, near freight depots, nearly new, all of brick. Address City Bank, Battle Creek, Mich.

FOR SALE—One Laidlaw Dunn Gordon Co. single-stage air compressor in good running order, size 14″x16″x18″. Address The Cincinnati Ice Co., Race and Canal streets, Cincinnati, O.

MACHINERY FOR SALE — Two 20-ton swing cranes, about 30 feet radius and 23 feet lift; hoisting, lowering and trolley movement by both power and hand drive. These cranes may be seen in use at Southwark Foundry & Machine Co., 5th St. and Washington Ave., Philadelphia, or drawing will be sent on application.

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THE STEAM PLANT OF THE WOOD WORSTED MILL

The Relation of Heat Used for Power Production to that Used for Heating and Mill Processes—Low First Cost and High Ultimate Efficiency—Simplicity the Key-note—Would Greater Elaboration Have Paid?

Extending for over a quarter of a mile along the south bank of the Merrimac river at Lawrence, Mass., and six stories in hight, the new Wood mill of the American Woolen Company is the largest textile mill ever projected and built at one time. Interesting as would be a general description of this immense establishment with its escalator or traveling staircase and other up-to-date systems for increasing the output and the well-being of its employees, our attention must be confined to the steam plant. This portion of the installation is characterized by its extreme simplicity and low first cost combined with high efficiency.

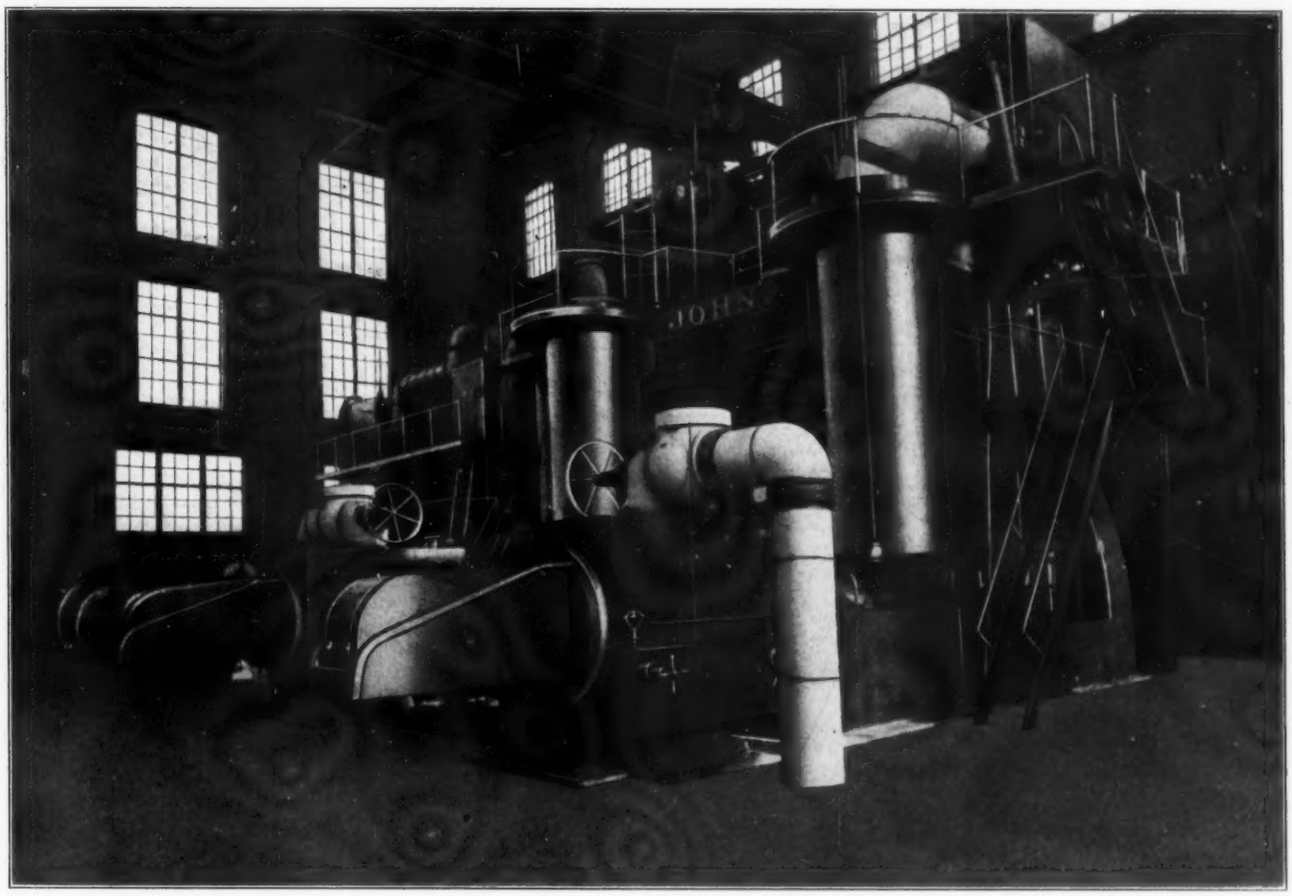


FIG. I. VIEW IN ENGINE-ROOM

In a textile mill the power problem is intimately connected with that of supplying steam for heating and manufacturing processes. In a woolen mill especially, enormous quantities of hot water are required, and since steam must be furnished for these purposes, large amounts of power may be generated at little extra cost by making the steam at high pressure, carrying it through the engines and exhausting it under sufficient pressure to serve the purposes of the mill, the engine serving in effect as a reducing valve and taking from the steam only the heat which it converts into power, which is thus obtained at great efficiency. The design of the plant is complicated by the necessity of adapting the engines, not to the demand for power which is reasonably constant, but to the varying demand for low-pressure steam, while furnishing an uninterrupted power supply.

On account of the large investment involved in carding and combing machinery it is desirable to run these departments continuously. To equip a mill with a carding and combing department large enough to keep the spindles and looms busy were it run only an equal number of hours, would involve an enormous investment. For this reason these departments of the Wood mill are designed to operate day and night.

The scheme of the power plant in brief is eight thousand boiler horse-power, an 1800-horse-power simple engine, and a 6000-horse-power compound unit. These are the rated powers at maximum steam efficiency, but where steam is needed for manufacturing purposes, steam efficiency is no object and the smaller engine can easily and has already developed continuously 2600 horse-power, while the larger will carry easily and continuously 8000. Both engines are designed for an initial pressure of 160 pounds. The smaller is run normally non-condensing and exhausts directly into the low-pressure steam system supplying the wash- and dye-houses, etc. The larger engine is compound condensing and its receivers are connected to the low-pressure system, furnishing enough low-pressure steam to

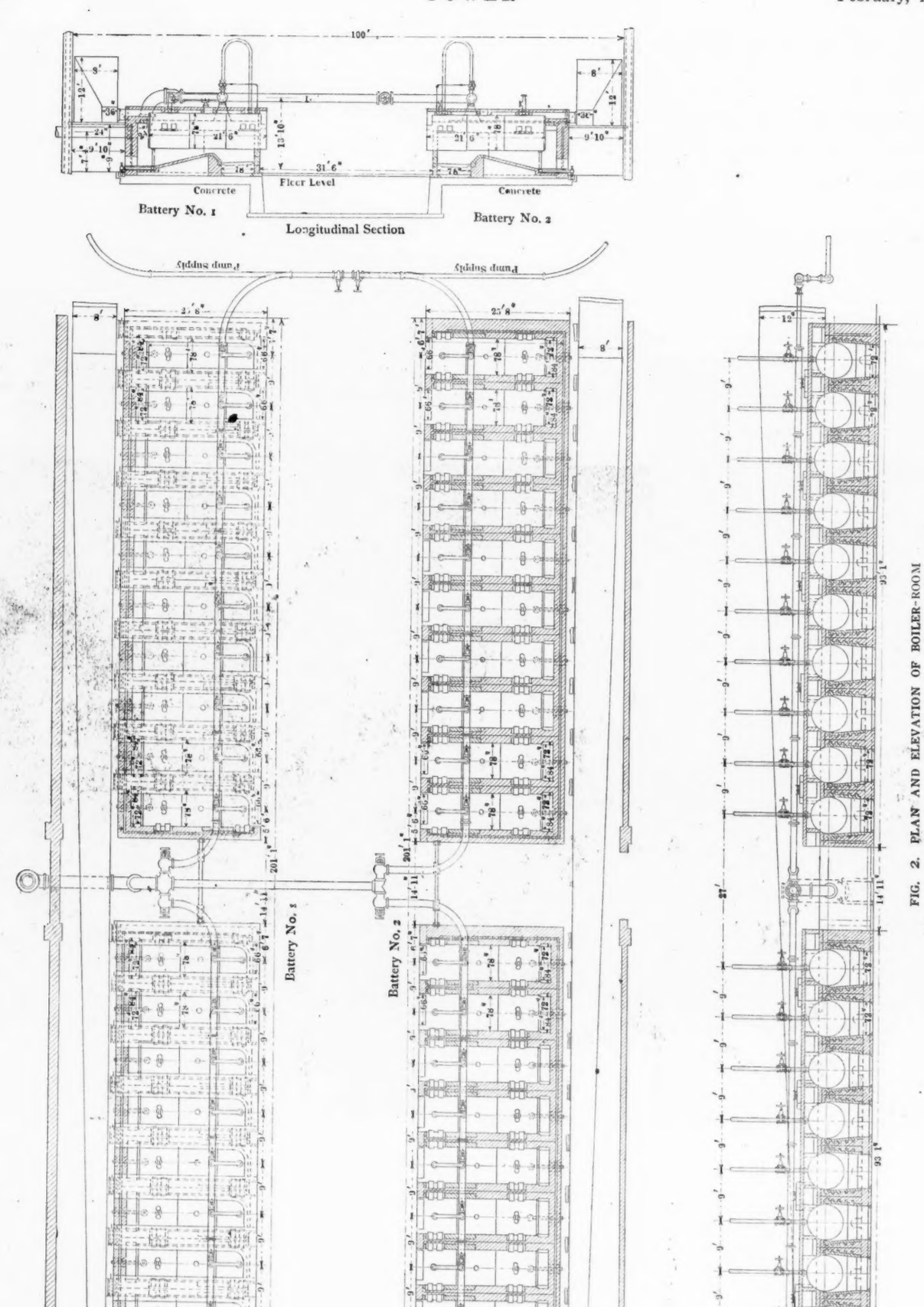


FIG. 2. PLAN AND ELEVATION OF BOILER-ROOM

make up the demand, while the rest is worked through the low-pressure cylinders to the surface condenser. Even here it is not rejected. The circulating water, uncontaminated by contact with the oily steam, goes to the dye-house and wet-finishing room, while the condensed steam is sent to the wool-scouring department where the cylinder-oil which it carries has no ill effect. All drips and auxiliary exhausts are carried into the same systems so that every heat unit furnished by the boilers is usefully applied and the only avenue of escape for unapplied heat units is in the flue gases, the temperature of which is reduced to the lowest possible point.

When the question of the steam plant was taken up George H. Diman, one of the engineers of the company, assured the management that if allowed to build the plant in his own way he would put it in for less than one-half of the first estimate, and was authorized by the president, William F. Wood, and Benjamin Smith, Jr., treasurer and general manager, to proceed. At present, but half of the boilers have been installed; an option for the duplication of the order was secured and doubling the cost of the present boiler instalment and including every conceivable item, the cost of the completed plant, including buildings, foundations, piping auxiliaries, and everything about the power plant but the electrical equipment, will not equal by $40,000 the amount he estimated. What he got for the money will be seen by the following description:

THE BOILER HOUSE

is of brick 328x108 feet, with roof trusses spanning the entire width and with a central monitor extending nearly the entire length. The forty boilers are set in four batteries of 10 boilers each, leaving a firing room of 31 feet 6 inches between the two batteries on either side and an alley 9 feet 10 inches in width behind each row as shown in the plan and elevation, Fig. 2, on page 74.

The boilers are horizontal tubulars 78 inches in diameter and 20 feet between tube sheets, containing each 143 three-inch tubes making a travel of 33.5 feet for the gases in contact with the heating surface after passing the bridgewall. The setting, which is somewhat of a departure, is the joint production of Messrs. Diman and F. B. Allen, chief inspector of the Hartford Steam Boiler Inspection and Insurance Company. The idea in view was to get away from the objectionable front flue and to keep the top of the boilers clean and clear.

The setting adopted is shown in Fig. 4. The gases pass over the bridgewall, along the bottom of the boiler, back through the tubes and then into the rectangular flue (above and at the right) with which each boiler is supplied and which discharges into a sheet-iron flue of gradually increasing section at the back as shown in the drawings. This flue extends over the alley back of the boilers, its bottom being nine feet from the floor level. At the boiler the most remote from the chimney it is 36 inches square, and the large end where the gases are delivered to the economizer is 8 feet in width by 12 feet high. The length of the flue is some 185 feet and in order to provide for expansion it is divided into 14-foot sections each resting upon a 3-inch iron pipe built into the walls of the building and the rear wall of the battery, and anchored thereto by angle-irons. These sections are connected by U-bends to allow for expansion. Across the brick partitions forming the individual flues, T-irons are laid supporting two courses of brick which form a smooth level floor over the entire battery as shown in Fig. 3. The boilers between the flue walls are covered with two inches of asbestos between which and the brick floor is a dead air space heated by the contiguous flue walls. As a consequence the radiation losses are very small and one can work on top of the battery without

FIG. 3. VIEW ON TOP OF BOILERS

discomfort. A thermometer laid by the writer on the brickwork immediately over the center of one of the boilers registered less than 100 degrees Fahrenheit after a long exposure.

The boilers are designed to carry 160 pounds pressure, built in three sheets of 7/16 steel having a tensile strength of 60,000 pounds, with butt-strapped, quadruple-riveted, longitudinal seams, and single-riveted lap roundabouts. The tubes are Shelby cold-drawn steel. The boilers are carried on two forged-steel lugs on each side, the rear lugs resting on plates and rolls in the usual way. The unusually heavy walls between the boilers will be recognized as resulting from the flue arrangement. Flat shaking grates 6 feet in width and 6 feet 6 inches long are used. Another peculiarity of the plant is the absence of ash-pit doors. It is a well-known fact that boiler makers use the same size of ash-pit door for all sizes of boilers. It is noticeable, too, that fires burn better right above the ash-pit door where the air enters. The fronts of these boilers are cut away for the full width of the furnace, insuring an equal distribution of air all over the grate and precluding the possibility of burning up grate bars by a closed ash-pit. It is bad policy to control the draft by the ash-pit doors; that is what the damper is for. Each boiler is provided with a damper in the connection to the main flue, but even this is not under the manipulation of the fireman. A draft gage is applied to each boiler, and this damper so adjusted as to give that boiler the same draft as all the others. It is then locked in position by the device shown in Fig. 11, on page 81, and as the speed of the draft-producing fan is varied to meet the varying demands for steam all of the boilers feel the change alike.

The alternate system of firing is used, twenty-odd pounds of semi-bituminous coal being burned per hour per square foot of grate surface, with scarcely a noticeable output at the top of the stack. A time system of firing such as is used in the United States Navy will be adopted. A red incandescent lamp is to be mounted above each firing door and lighted automatically at regular intervals, when that furnace should be fired. The boilers are supplied with approved types of fittings including high- and low-water alarms and two 4-inch pop safety valves. They have 2300 square feet of heating surface each, and would thus rate at about 200 horsepower apiece with 15 pounds of coal per square foot of grate, but with the higher rate of combustion here used will produce proportionately greater power.

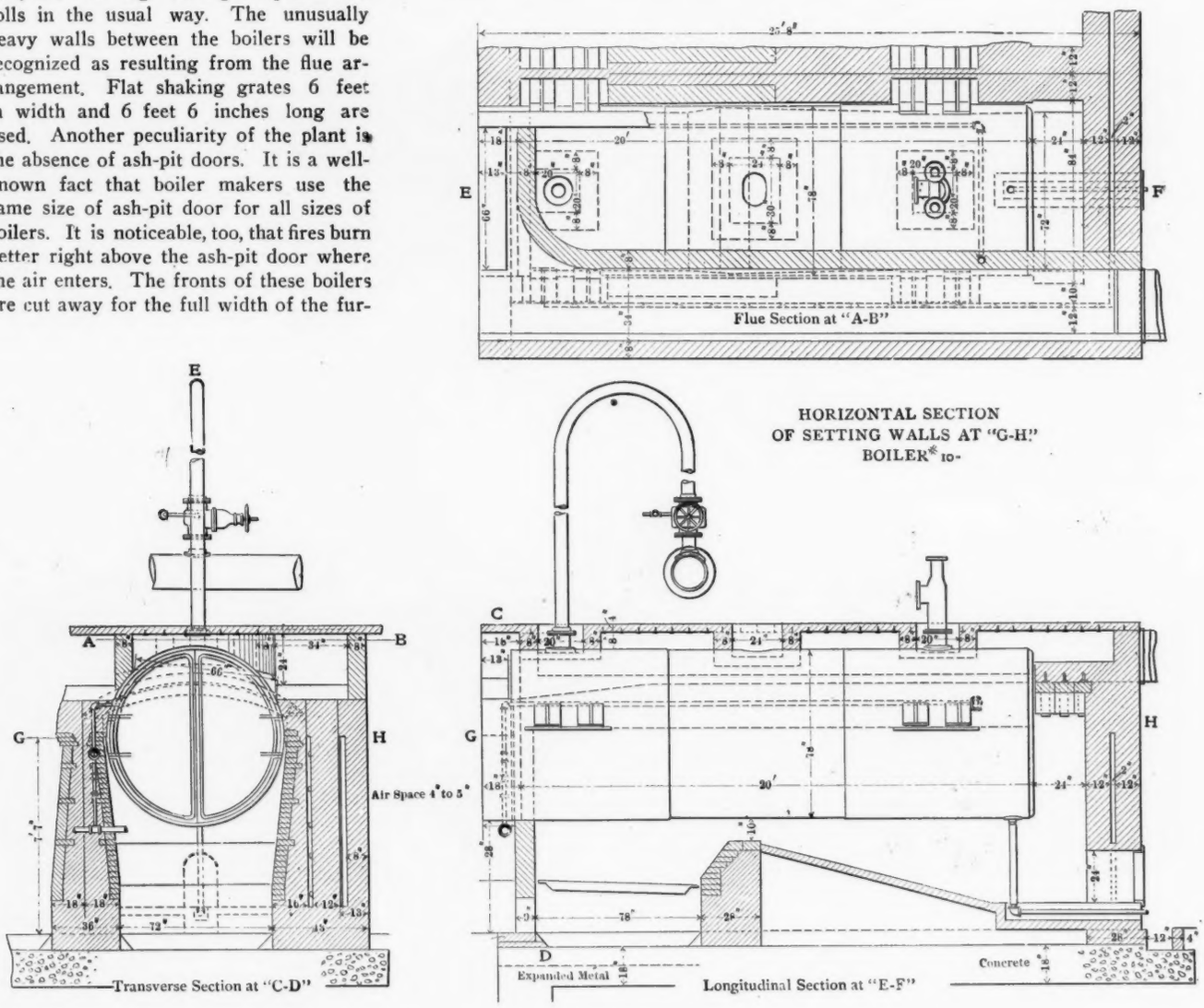


FIG. 4. BOILER SETTING

PIPING

The piping system is so simple that it can be readily comprehended from the drawings and photographs. The six-inch steam nozzles are situated at the front of the boilers, and are connected by U-bends to the 14-inch main which lies in cradles resting upon steel rolls which in turn rest upon cast-iron plates supported by brick piers immediately over the flue walls. Extra-heavy gate valves are used immediately above the main and dripped into the boilers at the front, the communication to the top of the combination columns being taken from a T in the drip-pipe. The safety valves of each boiler are set on a Y provided with a side outlet looking to the rear of the boiler and blank-flanged, furnishing numerous possible outlets for steam which may be needed without cutting into the main. All steam nozzles are forged, and are riveted onto the main, which has Van Stone flanges and joints, and no cast-iron fittings are used. The two 14-inch mains join in a 24-inch header through 14-inch angle valves, as shown in the plan Fig. 2, connected by a T on battery No. 2, and by a cross on the first battery, the smaller side outlet of which cross will receive the 14-inch main

FIG. 5. VIEW IN BOILER-ROOM LOOKING NORTH

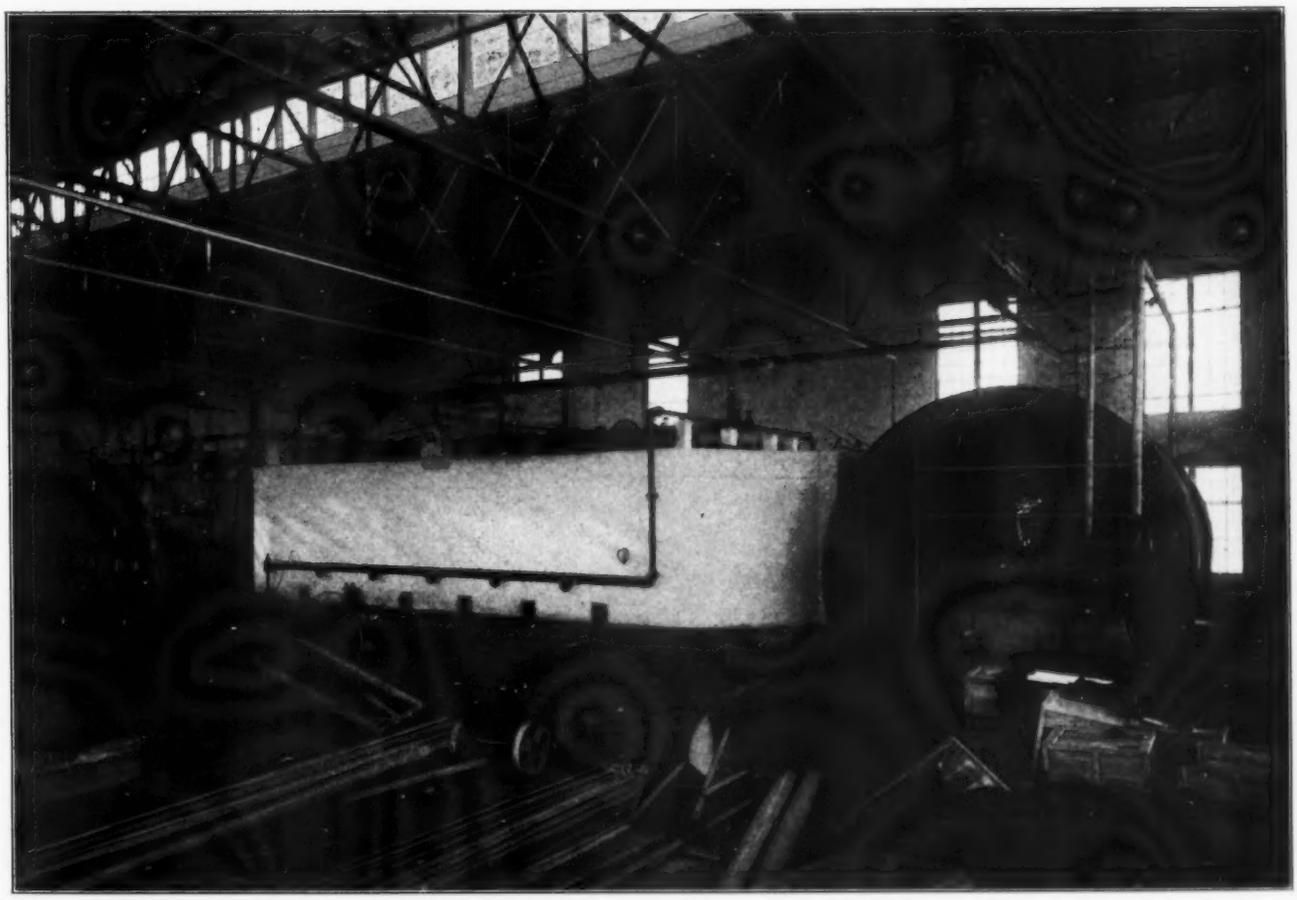


FIG. 6. ECONOMIZER AND FANS

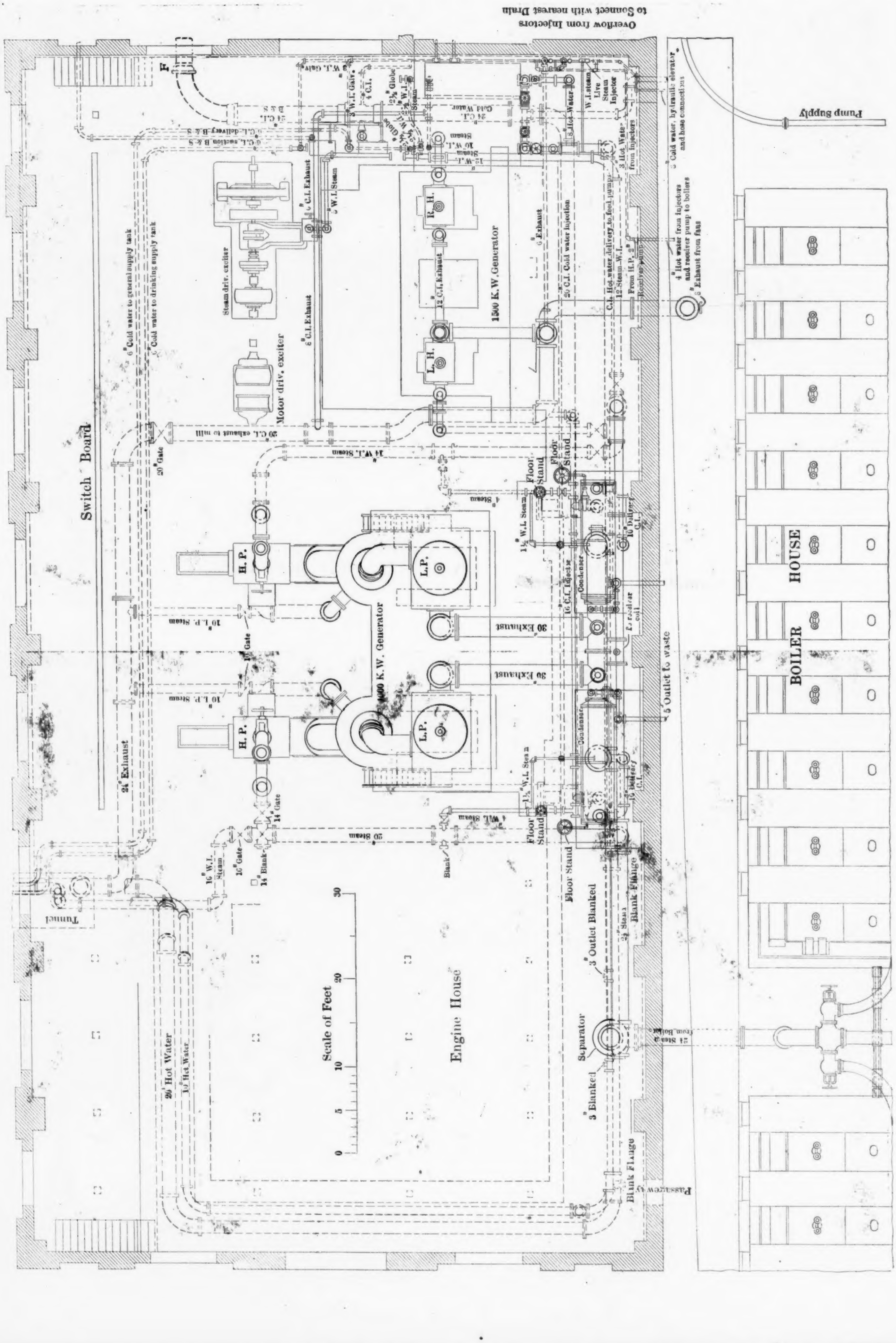


FIG. 7. PLAN OF ENGINE-ROOM

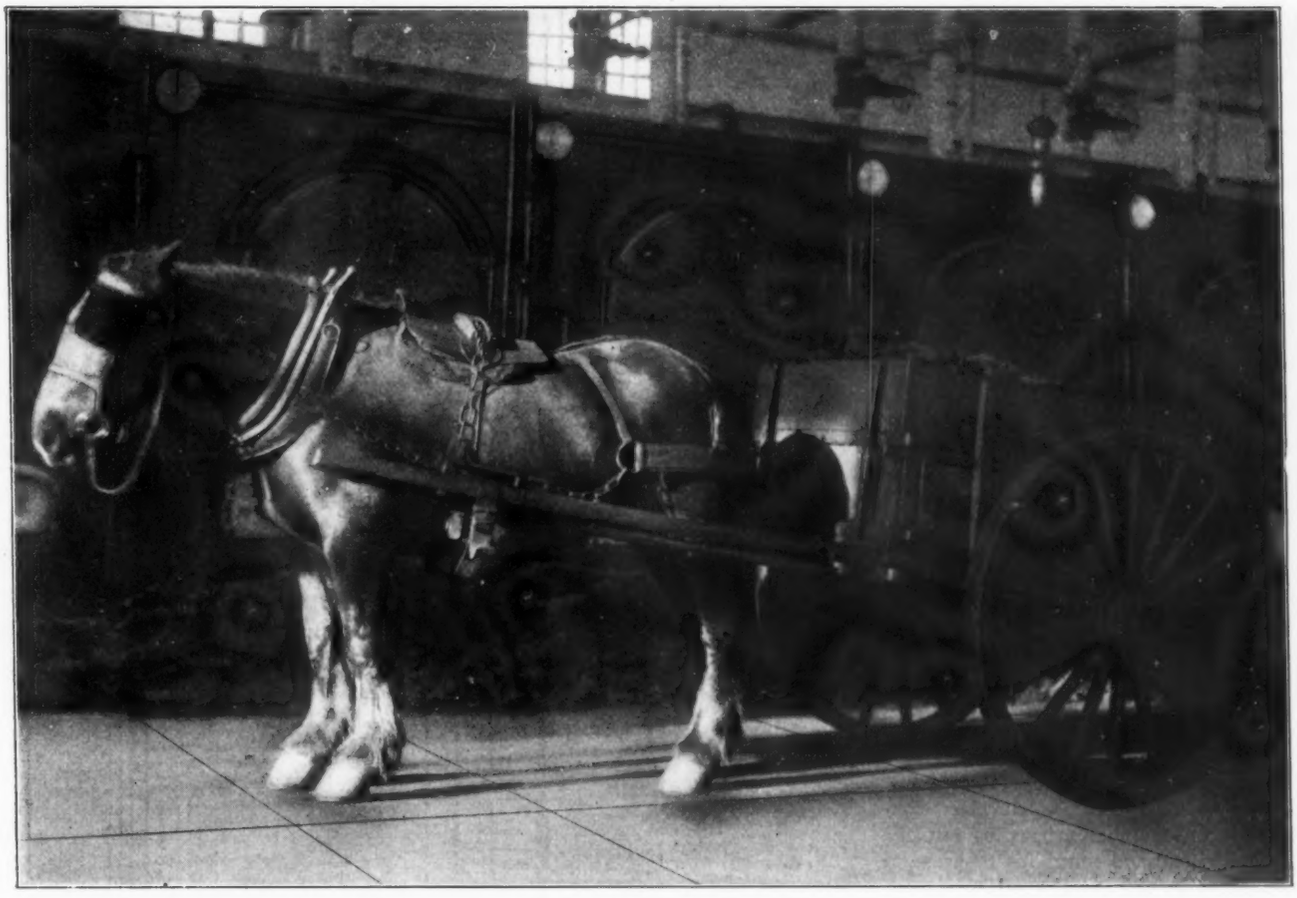


FIG. 8. THE COAL CONVEYER

FIG. 9. THE ASH CONVEYER

from the other battery. The layout of the piping is such that it is entirely free to expand, and it is not anchored at any point between the boiler outlets and the receiver in the basement of the engine-room, which the 24-inch main reaches by turning downward and passing under the flue.

DRAFT APPARATUS

As has been said, the only outlet for unapplied heat is by way of the chimney, and it has been the purpose of the designer to make this flow as small as possible. By the use of an exceptionally long boiler, and a liberal ratio of heating to grate surface, some 59:1, the gases are reduced to about 450 degrees as delivered to the economizer, and by it reduced to about 220 degrees before being delivered to the stack. With this low temperature a chimney which would produce the necessary intensity of draft would be impracticable. On the other hand, with so large a demand for exhaust steam, auxiliaries are run very cheaply. The draft for each side of the house is therefore supplied by two 15-foot fans working on the induced system between the economizer and the chimney where they receive the gases at their lowest temperature. The fans are operated by small direct-connected steam engines exhausting into the heating system and controlled by pressure regulators, so that as the pressure tends to fall the fan speeds up, increasing the intensity of the draft.

Either of these fans is of sufficient capacity when crowded to take care of the twenty boilers discharging into each economizer; so that, while both are used continuously, no interruption to the work would result from the failure of one, each being guaranteed to furnish an inch and a quarter of draft at the boiler most remote from the stack. Each side of the boiler-house has a self-supporting steel stack 120 feet in hight, and 10 feet inside diameter, each stack discharging the gases from twenty boilers.

FEED-WATER

The feed-water supply is taken from the circulating-water discharge from the condensers at a temperature of about 110 degrees, and in order to absorb as much heat as possible in the economizer no other heaters are used, the above temperature being ample to prevent the tubes from sweating. The water is pumped to the economizers under pressure and passes from them to the boilers at about 195 degrees. Exhaust-steam injectors are used instead of the boiler-feed pumps when the condenser is not running, heating the water before it passes to the economizer as well as furnishing the energy to put it into the boilers.

The economizers contain 1600 tubes, each tube having a little over ten square feet of heating surface.

COAL- AND ASH-CONVEYING MACHINERY

Mr. Diman believes that the simplest and most flexible means for handling coal

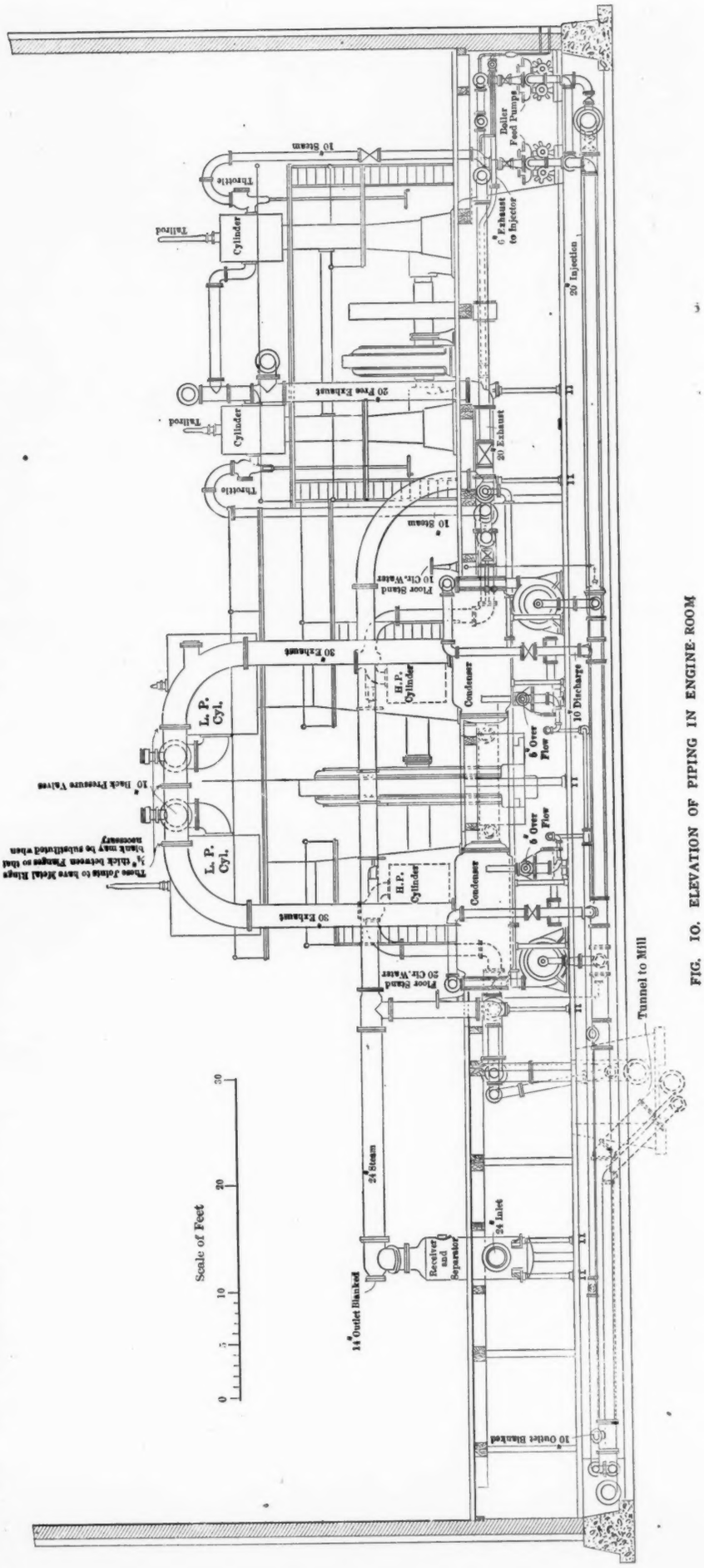


FIG. 10. ELEVATION OF PIPING IN ENGINE-ROOM

from the coal pocket to the furnace is animal muscle. A tip-cart with a capacity of one ton brings the coal to the firing floor and the firemen shovel it onto the grate. Four men at the coal pocket, one man to drive the two teams, one of which is being filled while the other is gone with its load, and a fireman to each five fire doors does the business, with a minimum of investment and no cost for upkeep and repairs. The cost of conveying the coal from the pocket, where it is delivered by the railroad, to the boiler-room floor is about $3000 a year, which Mr. Diman contends would not pay interest, depreciation, taxes, insurance, operation, repair and renewal charges on a conveying system and the additional cost of a building adequate to accommodate an elevated coal pocket. The original intention to have the railway spur run over the boiler house and drop the coal directly into an elevated pocket, had to be abandoned on account of the impracticability of the grade. The ashes are pulled out into openings in the boiler-room floor, ordinarily closed with

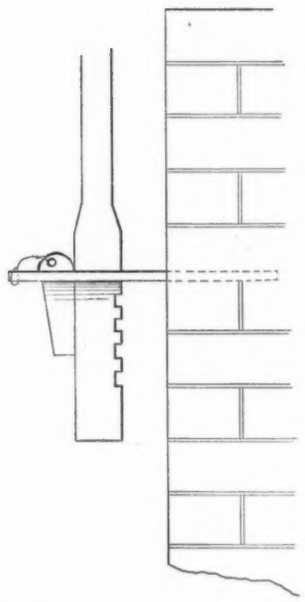


FIG. 11. LOCKING DEVICE FOR DAMPER

iron plates, and fall into the basement, whence they are carted off in a wheelbarrow, and raised by a hydraulic elevator to a level from which they can be loaded into wagons or cars by gravity, one man taking care of the ashes from twenty boilers.

THE ENGINE HOUSE

is 123x75 feet with a clear rise of 50 feet from the floor to the lower member of the roof trusses with sanitary flooring, non-combustible, not slippery and impervious to water and oil. Its color is black with a stripe of red running across the room at each bay. The room contains at present a pair of simple verticals running a direct-connected alternator of 1300-kilowatts capacity and a double compound with horizontal high- and vertical low-pressure cylinders running a 4000-kilowatt direct-connected alternator. The power for operating the mills is generated in this room and distributed electrically, fifty motors ranging from 8 to 175 horsepower being already installed. There are through the mill also a number of small

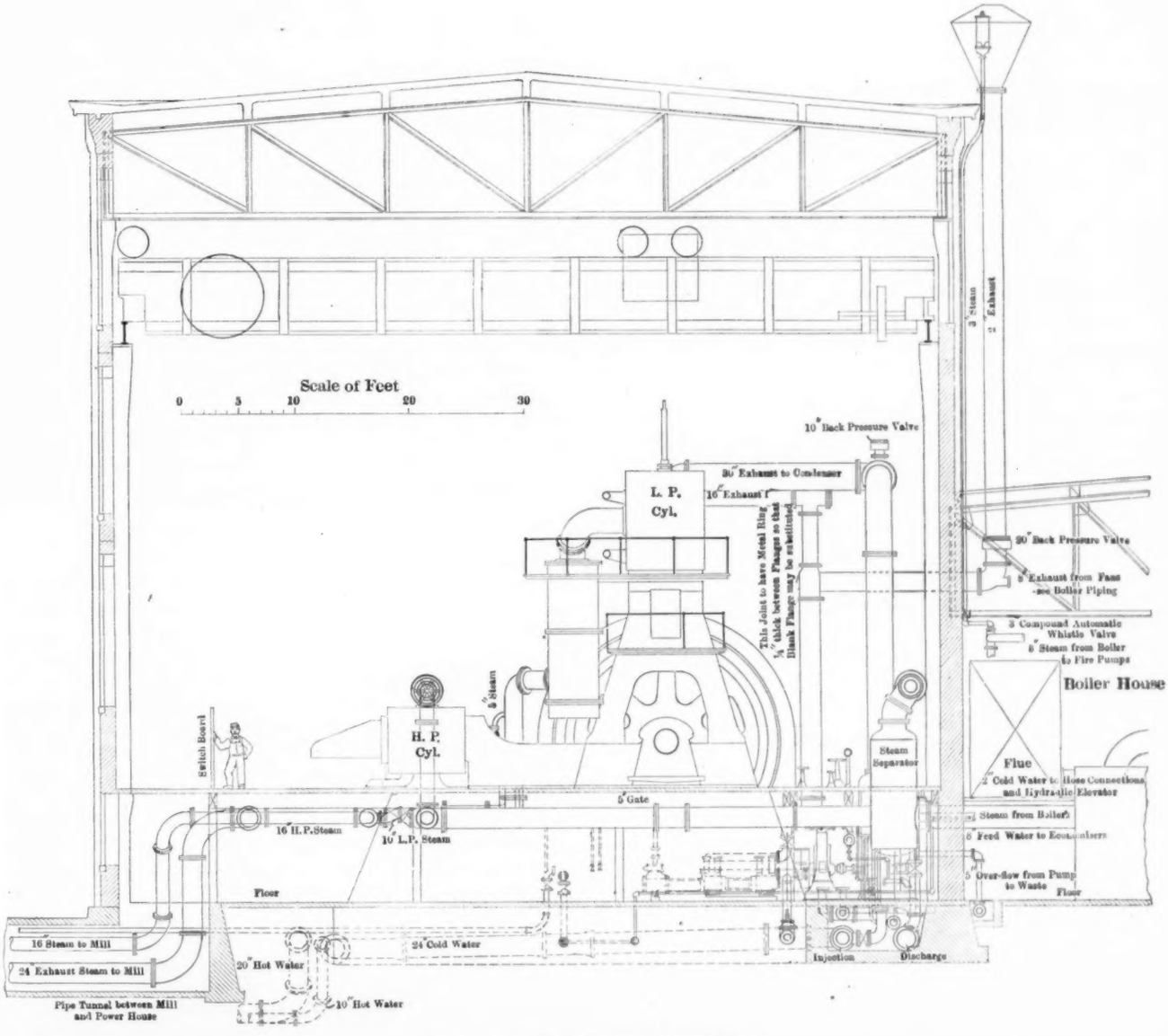


FIG. 12. ELEVATION IN ENGINE-ROOM

engines, driving the fans of the heating system, etc., the exhaust from which can be profitably turned into the low-pressure system. In the main engine-room there are also a steam-driven and motor-driven exciter and space is available for another main unit of the larger size. A 25-ton crane spans the building and covers the entire floor area.

McIntosh & Seymour engines are used for both main units. The smaller is 26x42 running at 133 revolutions per minute, guaranteed to develop 1600 horse-power cutting off at 0.22 and 2400 cutting off at 0.6 of the stroke, and has already developed 2600. The distance from center to center of the main bearings is 13 feet 1½ inches. It has a fly-wheel 14 feet in diameter carried on a 20-inch shaft, 17 inches in the bearings, which are 35 inches long; the length between bearing centers being about 12 feet. The crank-pin is 9 inches in diameter and 8 inches long, the crosshead-pin 8x8, the piston-rod 5 inches in diameter. Each cylinder has a 5-inch tail-rod.

The larger unit has 33-inch horizontal high-pressure and 68-inch vertical low-pressure cylinders giving a ratio of 4.25 and a stroke of 48 inches, which at its rated speed of 120 revolutions gives a piston speed of 960 feet per minute; distance from center to center of main bearings, 11 feet 11 inches. It is guaranteed to develop 5000 horse-power cutting off at 0.25 and 7100 at 0.5 of the stroke; one side has already developed 4160. It has no fly-wheel beside the generator which is carried on a 24-inch shaft, also 12 feet long between the centers of the bearings which are 22x42 inches. The crank-pins are 15x15, the crosshead-pins 9½ inches in diameter by 10½ inches long. The high-pressure rod is 7 and the low-pressure 6 inches in diameter. All four cylinders have tail-rods, those of the horizontal cylinders being seven inches in diameter cambered sufficiently to require the weight of the pistons to straighten them. The tail-rods of the vertical cylinders are 5.5 inches in diameter and terminate in eyes which furnish a hold for the crane when it is desired to remove the piston. The smaller unit was named "Priscilla" in honor of Mr. Diman's little granddaughter who started it, and the larger was appropriately christened "John Alden" by Mr. Wood.

THE ENGINE-ROOM PIPING

is shown in plan in Fig. 7, and in elevation in Figs. 10 and 12. In Fig. 10, which looks at the system from the boiler-room side, steam from the boiler-room enters the receiver at *A* and passes therefrom by a 24-inch main to a reducing T with 20-inch outlets. The downwardly extending branch turns horizontally under the floor and continues as shown in the plan Fig. 7 to the line of the high-pressure cylinder of the larger unit where the 14-inch steam pipe to the right-hand cylinder is taken off and the main, reduced to 16 inches, continues into the mill to furnish live steam for running the blowing engines, etc., and supplying live steam where the exhaust is insufficient. The horizontal branch sweeps, by a curve of long radius, as shown in Fig. 10, to a 20x12x14-inch tee beneath the floor, from which lead the supplies for the left-hand high-pressure cylinder of the larger unit and both cylinders of the smaller, as shown on the plan.

THE CONDENSERS

The condensing outfit is divided into two units with the object first of avoiding the practical difficulties of operating an oversized condenser under light loads and secondly of having one condenser capable of holding most of the load, with perhaps an impaired vacuum, in case the other fails. Each condenser has a cooling surface of 4500 square feet. The circulating water is handled by centrifugal and the condensed steam by reciprocating pumps, both steam-driven. The condensed steam at 110 degrees is taken by a small pump cylinder attached to the air pump, its piston-rod being a prolongation of that of the air pump itself, and pumped against a head of 40 feet to a tank in the wool-washing room. The circulating water at about the same temperature is discharged from the condenser into a main which conducts it to a concrete tank of 100 tons capacity in the dye-house, the centrifugal pump being proportioned to furnish the necessary head. These hot-water mains will be seen extending around the left-hand end of the basement in the plan Fig. 7.

In order to avoid the expense and leakage of numerous large valves in the exhaust system many of the flanges are made up with ¼-inch packing rings, so that blanks of quarter-inch steel, which are kept on hand drilled to fit the bolt holes and with projecting tags whereby they may be handled, may be inserted when at rare intervals it is necessary to stop a pipe off. In order to avoid a shut-down the 4-inch delivery of each condenser is provided with a valve the closure of which would seal the system and allow the plant to keep on operating with the other condenser until the end of the run, when the blank flanges would be inserted. The smaller unit is provided with an outboard or atmospheric exhaust, which serves really as a back-pressure or relief valve for the whole low-pressure system. Since the receivers of the compound unit are connected to this system its exhaust pipe has no atmospheric branch, as in case of the failure of both condensers the engine would be run high pressure and exhaust into the low-pressure main. To preclude dangerous accumulations of pressure in the exhaust-pipe system from any cause, however, each side is provided with a 10-inch relief valve as shown in Fig. 10.

All of the exhausts from the auxiliaries are led into the 20-inch cast-iron exhaust main which becomes 24-inch beyond the large engine and is carried in that size into the mill. Under normal working conditions a pressure of six pounds will be maintained in the system. At *E* and *F*, Fig. 7, are shown ten-inch connections to this main from the exhaust pipes of the high-pressure cylinders of the compound unit, allowing as much steam to be bled from the receivers as is required to maintain this pressure, the rest going into the low-pressure cylinders. This accounts for the fact that there is so much engine for the size of the generator in this unit, the low-pressure portion being liable at times to do only a small amount of work, but this will be at the time when the demand for exhaust steam is pressing and the high-pressure cylinders must furnish it. A connection is also made to this main from the 16-inch live-steam main at *G*, through a reducing valve, so that live steam may pass in and maintain the pressure when the supply of exhaust is insufficient.

The plant has certainly been put down at a remarkably low first cost, and it is difficult to see where its efficiency for the purposes for which it was designed could have been improved. Nobody questions the efficiency of the type of boiler adopted, if one has the space for them, which was the case in this instance; and the engine equipment is first class and well adapted to the service. The buildings are substantial and in keeping with the high character of the rest of the plant, ample in size, lofty, airy and well lighted and the piping and auxiliaries are of the best. The only place where further investment might save anything would be in labor cost of operation as by the use of mechanical coal conveyers and stokers, and Mr. Diman would like to be shown how his plant can be bettered in this respect.

Largest Producer-gas Power Plant in the World

The new plant of the Iola Portland Cement Company at Dallas, Texas, will have the largest producer-gas power installation in the world. The Loomis-Pettibone pressure producer system, built by the Power and Mining Machinery Company, will be furnished in three units having a total capacity of 4500 horse-power. Bituminous coal and Texas lignite will be the fuels used.

The Snow Steam Pump Works, of Buffalo, will furnish for this plant four single-tandem double-acting gas engines, each with a normal capacity of 1100 brake horse-power. These engines will be direct connected to alternating-current 25-cycle electric generators of 810 kilowatts capacity each, operated in parallel.

Continuous operation, twenty-four hours per day, seven days per week, will be required of the plant.

ELEMENTARY LECTURES ON ELECTRICAL ENGINEERING—XIII

The Direct-current Motor—Why Its Armature Revolves

If a direct-current dynamo be connected to a source of electric current of the same voltage as that of the dynamo, the armature of the machine will revolve at almost the speed at which it was designed to be driven. The machine then becomes a motor, capable of driving other machinery. This reversibility of the dynamo is due to the fact that if two magnetic fields be set up near each other they will tend to arrange themselves so that their magnetic lines of force agree. For example, suppose two coils of wire be arranged as shown in Fig. 34, the one, *F*, secured to some stationary support and the other one, *A*, hung on pivots. If a current be passed through each of them in such relative directions that the magnetic fields have the polarities indicated by the arrows in the sketch, the coil *A* will swing over until the two are in line, or as nearly so as the fastenings of the coils will allow, and the two fields will combine, as indicated in Fig. 35. These elementary sketches also illustrate the fundamental feature of magnetic force, namely, the tendency of magnetic lines or flux to reduce the length of the path to the shortest possible distance. The two fields set up by the two coils could combine partially even with the coil *A* in the oblique position represented in Fig. 34, and if the coil were immovable there would be a partial combination of

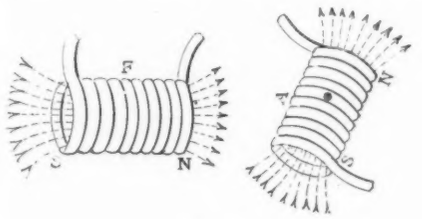


FIG. 34

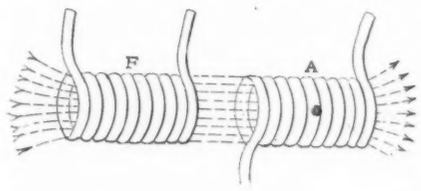


FIG. 35

the two fields. But the path of the combined field would be much longer than in Fig. 35, and it is the effort of the magnetic lines to shorten this path that swings the coil *A* into the straight-line position.

The reaction of a motor armature upon its field magnet is similar to that between the two coils just explained. If the armature of a bipolar motor were blocked so it could not turn and current were allowed to flow through it by way of the brushes, it would be found that in all of the wires lying under one pole-face and in those near the edges of the pole-face on each side the current would be flowing from the front to the back end of the armature, while in the wires under and near the other pole-face the current would be flowing in the opposite direction. In a few wires located diametrically opposite each other and at points almost midway between poles there would be found no current at all. This condition is indicated roughly by Fig. 36, where the circles containing crosses represent wires in which the current is flowing away from the reader and those containing dots represent wires in which the current is flowing toward the reader. The two clear circles represent wires in which no current is flowing, these wires being short-circuited by the brushes on the commutator (not shown). The line *c*—*c* indicates the "neutral axis" or axis of commutation and it is also the "polar axis" of the winding. That is, if the armature be considered as a magnet (any magnetic material, whatever its shape, will be a magnet if surrounded by a current-carrying wire, or symmetrical group of wires), this line will be its magnetic axis.

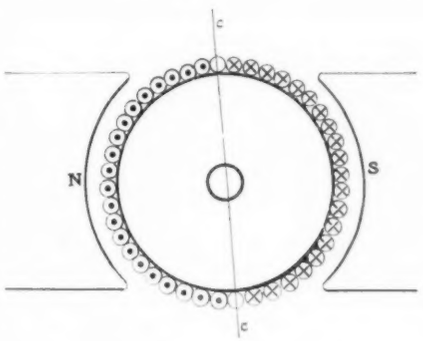


FIG. 36. CURRENT IN ARMATURE CONDUCTORS

With current flowing in the armature winding as indicated by Fig. 36, the effect will be the same as though the wires formed a simple helix, as in Fig. 37; that is, a magnetic flux will be set up in the armature core in the direction indicated by the dotted lines and arrowheads. This flux is nearly at right angles to the flux in the field magnet, which is indicated by the short arrows at the pole-faces, and the result of this angular displacement is an effort by both fluxes to get into alinement. This effort distorts the field flux from its straight path across the armature core and distorts the armature flux to meet it, as indicated crudely by the arrows in Fig. 38, these arrows representing the axes of the two fluxes. Both fluxes exert an effort to straighten out, and as the armature core is movable, the fluxes turn it in the direction necessary to bring them into line; that is, clockwise. If the current passed in and out of the armature winding through two direct terminals, as in Fig. 37, the core would turn over until

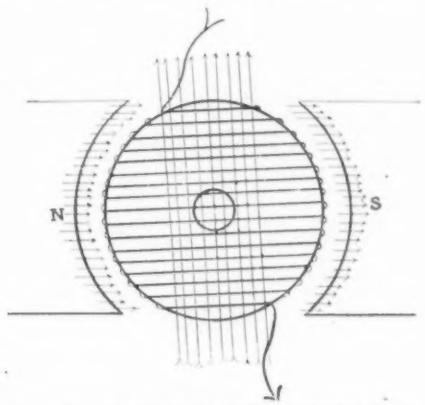


FIG. 37. ELECTRO MAGNETIC EQUIVALENT OF FIG. 36

the line *c*—*c*, Fig. 36, became parallel with the center line of the two field-magnet poles, and then it would stand still, with the two fluxes perfectly in line with each other. But as soon as the armature turns far enough to let the brushes slip from the commutator bars on which they first rested to the next succeeding bars, the magnetic axis of the armature is shifted backward to where it was at first, and the effort to straighten the two fluxes begins all over again. The armature turns through another small arc and the brushes slip to the next commutator bars, shifting the polar axis of the armature back again, and so on and on, as long as current is supplied to the armature and field windings. The rotation of the armature is continuous—not step by step—be-

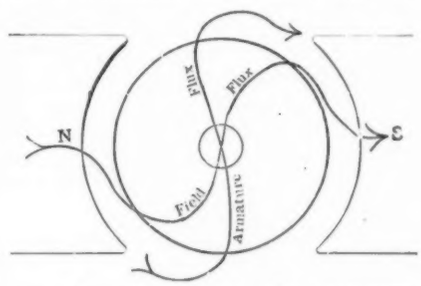


FIG. 38. FIELD-MAGNET AND ARMATURE FLUXES

cause the commutator bars are so close together that the brushes always touch them.

The shifting of the polar axis is illustrated by Figs. 39 and 40, which show an armature in two positions 1/16 of a revolution apart. For the sake of simplicity only six of the commutator bars

are shown connected to the winding and the full connections of the latter are not shown. In Fig. 39 the brushes rest on bars connected to the wires *b* and *y*, which are located in the polar axis (see Fig. 36). In Fig. 40 the armature has rotated 1/16 of a revolution in its effort to twist its polar axis into line with that of the field magnet, but the brushes are now in direct connection with the wires *c* and *z*, which occupy the positions formerly occupied by the wires *b* and *y*; the polar

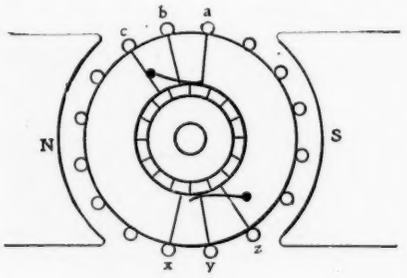


FIG. 39

axis is thereby set back to where it was before, because it always passes through the two wires with which the brushes are directly connected by the commutator. In the course of the travel from the position of Fig. 39 to that of Fig. 40, there is an instant when the upper brush touches the commutator bars of the wires *b* and *c* simultaneously and the lower brush similarly touches the bars connected to *y* and *z*; this short-circuits two of the armature coils during the brief movement while the double contact exists, and it is this short-

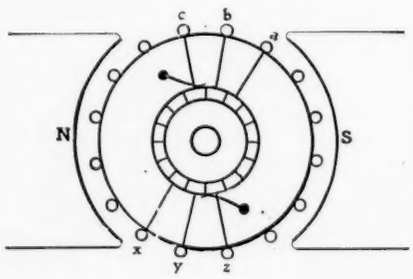


FIG. 40

circuiting that causes some of the wires on opposite sides of the armature to be without current in them, as indicated in Fig. 36.

In all of these sketches each wire represents an armature coil, but, as in the case of the dynamo, an actual armature coil may contain a number of wires.

In 1887 an engine designed by E. D. Leavitt and built by the Dixon Iron Works of Scranton, Penn., was put into the Washington Mills at Lawrence, Mass. This engine was described in detail in our issue of June, 1889. It is still in active operation and we are informed by the engineer that the same babbitt is running in the main bearings and the crank boxes, and in fact in all the bearings except those of the crosshead-pin, as was in the engine when it was first installed.

The Lynn Explosion

BY THOMAS HAWLEY, EX-CHIEF STATE INSPECTOR

On Thursday morning, December 6, 1906, at a few minutes before seven o'clock, the boiler in the factory of the P. J. Harney Shoe Company, Lynn, Mass., exploded, demolishing a portion of the building. The factory was a four-story wooden building, and the resulting fire completely destroyed it, together with the adjoining wooden factories of the Owen Paper Box Company and the Tufts & Friedman Shoe Company. The fire then spread across the tracks of the Boston & Maine Railroad, destroying the West Lynn station of that road, the Russell Box Company factory, the Gower shoe factory, and eleven small dwellings. As the explosion occurred before starting time, there were very few operatives in the buildings; still, at least fifteen persons were injured or burned, but none fatally. The fine photographs herewith excellently portray the wreck.

The exploded boiler was ten years old. It was of the horizontal tubular type, 72 inches in diameter and 18 feet long, made of "Clyde" fire-box steel, a scant ⅜ of an inch thick, built in three courses, with a lug on each course, the horizontal seams being of the lap double-riveted type. There were ninety-two 3½-inch tubes and the head was braced by eight 1¼-inch round through rods passing through channel bars, four braces in a row; there were also two 1¼-inch radial braces on each head forked to angle-irons riveted to the upper part of the head. The boiler was built by the Cunningham Iron Works, Boston, and was regularly inspected by the Hartford Steam Boiler Inspection and Insurance Company. The pressure ordinarily carried was about 90 pounds.

The boiler had been under a light fire during the previous night, in charge of a watchman who held a license as fireman. The engineer, also a licensed man, was in charge at the time of the explosion and was standing in the engine-room at the rear of the boiler house, getting ready to start the engine, although the throttle had not been opened. He escaped with some bruises and a general nervous shake-up. The engineer of the neighboring Owen Paper Box Company factory was cut about the head, the greater portion of the front course of the exploded boiler landing quite near him.

The damage by the explosion itself does not appear to have been very considerable, the greater loss resulting from the extensive conflagration produced by the inflammable character of the buildings and contents. The remains of the wrecked boiler reveal that the rear course, with the tubes in its head, was thrown back about eight feet. The second course lay flattened out where the boiler had stood, and the first course, also flattened out, with the front head partly attached, was blown about twenty feet toward the Owen factory. Examination of these plates indicated that the boiler parted at the lap-seam of the middle course, the break extending clear across the plate along the edge of the lap, but not through the rivet holes. The rupture then continued along both girth seams, tearing partly through the sheet, shearing some of the rivets, or breaking the plate between the girth rivet holes, and separating this plate from the others.

At the longitudinal seam of the front course the rupture continued along the line of these rivets for about eighteen inches, and a portion of this sheet remained attached to the second course. From the end of the fracture at the longitudinal seam of the front course, the fracture extended diagonally downward, tearing through the solid sheet to the head rivets, then around this row of rivets, partly shearing and partly breaking the plate through the rivet holes, for two-thirds of the distance, and leaving the dry sheet attached to the front head. This course had the safety-valve outlet, but the valve had not, at this writing, been found.

From the testimony of the engineer, as well as the engineer of the adjoining building, it seems reasonably sure that the safety valve was in working order, for it had been heard to blow several times the day before, and the engineer had observed, a few minutes before the explosion, that the pressure was well within the blowing-off point.

The accident is undoubtedly another of the many lap-seam explosions that have recently been recorded. This is the fourth very disastrous explosion in Massachusetts within a few years, known to be from this cause, the others being that at the Langley harness factory in Fall River, the Acushnet mills in New Bedford, and the Brockton explosion of last year—all very disastrous. The cause of the crack extending along the lap-seam has already been given in POWER (page 601, October, 1905, issue) and an examination of this boiler leads to no new conclusions. The crack being under the overlap of the plate could not be discovered by such inspection as could be given it, and although suspected in all large boilers of this type there is no direct evidence that it had shown itself prior to the explosion.

The plate itself has all the appearances of extreme brittleness; pieces could be broken off short along the edge of the fracture with a few sharp blows of a hammer, and the plate also indicated very little ability to stretch. At no place in the fracture, either along the rows of rivets or where the plate was torn across the solid plate, does there appear any reduction in area, usually found with ductile plates. The State inspectors have cut out portions of the plate to submit them to tensile and physical tests to determine their present character, and from the ease with which pieces of some size can be broken

FRONT COURSE; WHOLE HEAD PARTLY ATTACHED

Shows how fracture followed along lap-seam of front course, tearing through plate to head rivets, then tearing plate from head

PORTION OF **FRONT COURSE AND HEAD.** TO LEFT, REAR COURSE WITH TUBES

THE RUINS, LOOKING TOWARD THE RAILROAD

In front of State Inspectors Hinckley and Ferguson is flattened-out second course, at the horizontal seam of which fracture started

VIEW OF REAR COURSE AND CORLISS ENGINE NEAR WHICH ENGINEER WAS STANDING

off it is anticipated that these tests will show poor material.

The writer has made tests of this character, from test pieces cut out of exploded boilers as close to the original fracture as it was possible to get, yet in every case the test has shown ample tensile strength and an elongation of over 20 per cent., and it would not be surprising if these tests would result in the same way. These tests are of course no indication of the condition of the steel when put into the boiler, but indicate that test pieces can be cut out of a plate and submitted to the slow strain of a testing machine and show up very well; yet, when they fail, they show all the characteristics of a very bad boiler steel.

The problem of what to do with those large boilers of the lap-seam type remaining in use in Massachusetts will undoubtedly be taken up by the next legislature. It is a difficult thing to determine just what tests shall be made to discover these cracks, for the inspection companies do all they can, in such time as they can give for an inspection, to discover them, and it seems as though nothing short of positively prohibiting the use of these boilers will give absolute protection against them.

MIDDLE OR SECOND COURSE, WHERE RUPTURE STARTED

Shows where pieces have been cut out at left-hand end for tests.

Values of Fuels Produced in the United States in 1905

According to the figures compiled by the United States Geological Survey, for the fourth time in the history of the United States the production of coal in 1905 reached a total of over 300,000,000 short tons, showing an actual output of 392,919,341 tons of 2000 pounds, valued at $476,756,963. Of this total, the output of anthracite coal amounted to 69,339,152 long tons (equivalent to 77,659,850 short tons), which, as compared with the production of 65,318,490 long tons in 1904, was an increase of 4,020,662 long tons, or 6 per cent. The value of the anthracite coal at the mines in 1905 was $141,879,000, as against $138,974,020 in 1904.

The output of bituminous coal (which includes semi-anthracite and all semi-bituminous and lignite coals), amounted in 1905 to 315,259,491 short tons, valued at $334,877,963, as compared with 278,659,689 short tons, valued at $305,397,001, in 1904. The increase in the production of bituminous coal in 1905 over 1904 was therefore 36,599,802 short tons in quantity and $29,480,962 in value.

The coke production in the United States in 1905, which included the output from 3159 retort or by-product ovens, amounted to 32,231,129 short tons, as compared with 23,661,106 short tons in 1904. The increase in quantity in 1905 from 1904 was 8,570,023 short tons, or 36.22 per cent. The total value was $72,476,196, as against $46,144,941 in 1904, a gain of $26,331,255, or 57 per cent. The average price per ton in 1905 was $2.25, against $1.95 in 1904. The average output from the by-product ovens in 1905 was 1158.8 tons per oven, against an average of 365.8 tons per oven from the bee-hive ovens.

The aggregate value of all the products from the distillation of coal in gas works and retort ovens in 1905 was $56,684,972, as against $51,157,736 in 1904, and $47,830,600 in 1903.

An exchange says: The quantity of coal used by some of the big steamships is startling. The new turbine liner, Lusitania, has an indicated horse-power of 65,000, which means a consumption of 435 tons of steam per hour, which it will require 50 tons of coal to make, nearly a ton a minute. This means more than 1200 tons of coal a day, or in five days she will consume over 6000 tons of coal.

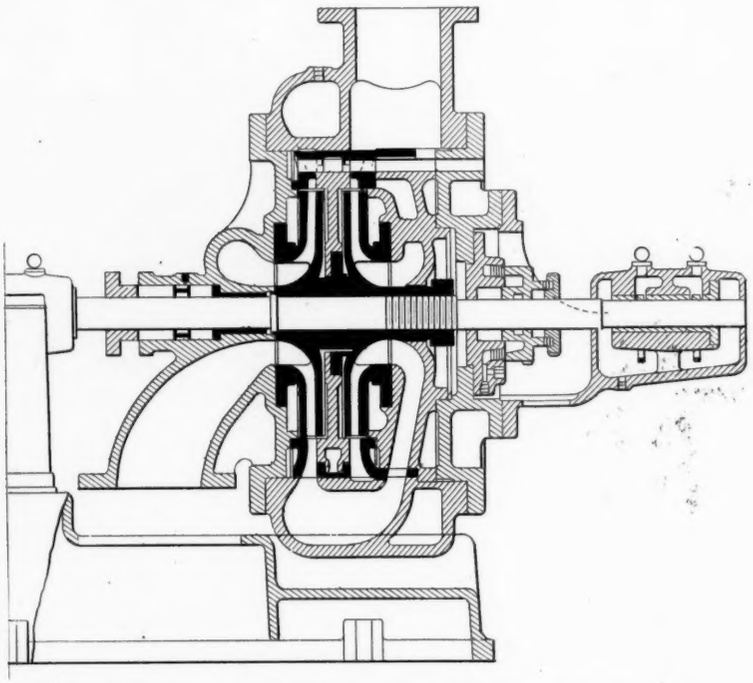
The Centrifugal Pump

CONTRIBUTED

Obviously the best way to consider centrifugal pumps is to compare them with their competitors, reciprocating pumps. In centrifugal pumps there is no shock or water hammer because the flow of the liquid is continuous and the rate uniform. The efficiency of the centrifugal pump is very nearly equal to that of the reciprocating type, especially when the centrifugal pump is direct connected to either an engine or a motor. Durability is another feature which contributes to the efficiency; after ten years of service the efficiency of the reciprocating pump, if equal expenditure for repairs were made on each machine, would be very low, whereas the efficiency of a centrifugal pump would be maintained because the only wearing parts are the bearings, which are lubricated with oil rings and require very infrequent replacement. Compare this with the wearing out of the water-cylinder linings of the reciprocating pump. Another important item is the oil bill. While the reciprocating pump demands cylinder lubrication, there is no corresponding expense connected with the operation of the centrifugal pump. Added to this is the ability of the centrifugal pump to handle gritty or acid liquids, and, as it has no valves, it may also handle water containing pulp or slime.

The speed of the centrifugal pump is so high that it may be direct connected to a motor or engine, whereas the reciprocating pump must be geared with one or two speed reductions, entailing loss of power and noise. Due to the higher speed, the centrifugal pump also occupies less space for a given capacity, and this is an important feature, especially in mine work.

Cenrifugal pumps may be divided into two distinct types, namely, the volute type and the turbine type. The volute type is appropriate for low heads and conditions where maximum efficiency is not absolutely necessary, while the turbine type is used under high heads and where maximum efficiency is required. In general, the turbine pump will require about ten per cent. less power for a given head and rate of water delivery than the volute pump under the same conditions, and while either type of pump may be built compound in two or more stages, it is almost universal practice to use only the turbine type for compound units; aside from a somewhat larger gain in efficiency, it lends itself more readily to compact mechanical arrangement.



LONGITUDINAL SECTION THROUGH A CENTRIFUGAL PUMP

All types of centrifugal pumps consist of two essential elements, namely, a rotating impeller which receives the water entering at the inlet, and a stationary water way in the outer shell, which receives the water thrown from the ends of the impeller blades. Upon the design of these two elements depends the success of the pump and the efficiency obtained. If the water were received from the inlet directly into the impeller without its course being changed by following a channel of the proper curvature and proportions, a large loss of power would result; secondly, if the water were discharged from the impeller directly into a large chamber, the efficiency of the pump would be very low, as the kinetic energy, represented by the velocity of the water, would be lost. The greater portion of this kinetic energy is utilized by directing the water into a throat and volute, or a diffusion ring.

The throat, or diffusion ring, is merely an expanding water way which allows the velocity of the water to be decreased gradually. The proper proportion of the throat and the correct curve for the vanes in the diffusion rings are the results of many tests covering several years. It may be stated without exception that the general form of the volute, or passage way, in which the water finally finds its way to the outlet of the pump, is that of an Archimedes spiral; that is, the cross-section increases in arithmetical ratio up to the full area of the pump discharge.

To extend the comparison between the direct-acting reciprocating pump and the centrifugal pump, consider the three chief variables, namely, head, rate of delivery and speed. Given any constant head, the delivery and speed are proportional, though not according to a simple function. A direct-acting pump, however, has a capacity proportional directly to the rate of speed. For any constant delivery rate the speed required is a function of the square root of the head, while if the speed be constant, the relation of delivered volume for a change of head, obtained with constant head, differs in that a greater change takes place in the delivered volume for a change of head, and also that the variations are in the opposite direction. This will be readily appreciated when it is remembered that the water horse-power must be approximately constant.

If v = the peripheral velocity of the runner in feet per second,

H = the head,

V = the velocity in feet per second of the water in the discharge pipe;

then

$$v = \sqrt{bH + aV^2} + aV$$

where a and b are constants,

$$H = 0.031\ K \times (v^2 - v \times V \times cot.\ \alpha),$$

in which K is a constant and α the angle of the blade at the circumference of the impeller.

If the water be gritty or contain solid matter of any nature, the usual speed cannot be employed owing to the excessive wear which would result if the pump were driven at as high a speed as with clear water. This is one of the most important considerations—the speed limitation of the pump—and is the factor which governs the capacity of dredging pumps. One way of providing for a lower rotative speed is to build the pump in two or more stages, thereby dividing the revolutions per minute by 2, 3, 4, etc. Although up to this time no pumps in this country have been constructed containing more than eight stages, is seems feasible to build them with more.

Catechism of Electricity—XIX

TRANSFORMERS

393. *What is a transformer?*

An apparatus used for changing the voltage of an alternating current.

394. *For what general purposes are transformers employed?*

To obtain low-voltage current from high-voltage supply circuits and to raise the voltage of alternating-current generators for delivery to transmission lines.

395. *Why is it desirable to obtain low-voltage current from high-voltage circuits?*

It is advantageous, in the first place, to operate transmission circuits at high voltage because this saves expense in the construction of the circuits by permitting the use of small wires. But the voltages at which line construction becomes economical are much too high for safe distribution to users of current; hence transformers are interposed to obtain current at safe pressures.

396. *Why is it necessary to use transformers to raise the voltage for transmission purposes?*

Because generators cannot be built to operate reliably at the high voltages required for long transmission lines. In order to keep the cost of very long lines within commercial limits the voltages are carried up as high as 60,000 volts, and generator armatures cannot yet be insulated to withstand any such pressure, or even 25,000 volts.

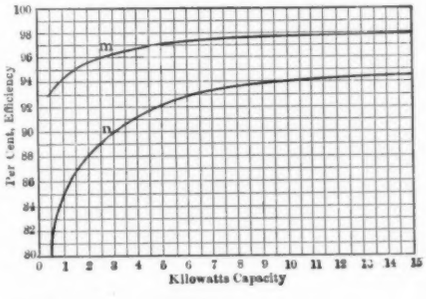


FIG. 84. EFFICIENCY CURVES OF A TRANSFORMER

397. *Are no names given to transformers to distinguish those that increase voltages from those that are used to decrease them?*

Yes; the former are called step-up transformers and the latter step-down transformers. Referring to Fig. 83, the step-up transformer *b* raises the voltage supplied to it by the alternator *a* for transmission over the line *c e*. The step-down transformer *s* lowers the line voltage for use in lamps, motors, etc., connected to the circuit *m*.

398. *Describe the general construction and the principles of operation of the transformer.*

A transformer consists essentially of two windings so placed on a soft iron core that the inductive action of the one upon the other is very great. Taking, for example, the transformer *b*, Fig. 83, one of the windings *n* is joined to the source of power and is called the primary winding. The alternating current flowing through this winding produces an alternating flux or flow of magnetic lines of force in the iron ring *u*, and these cutting the other winding *r*, which is called the secondary winding, induce in it an alternating electromotive force.

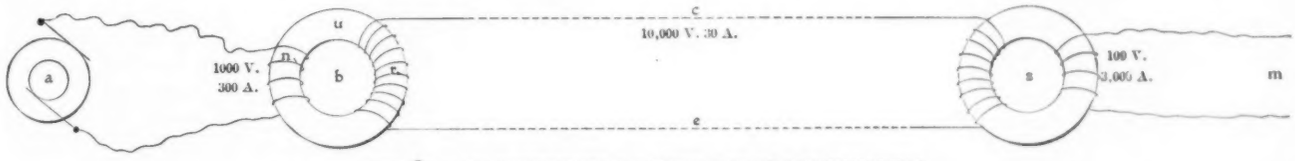


FIG. 83. AN ILLUSTRATION OF THE USE OF TRANSFORMERS

399. *Is there any fixed relation between the voltage applied to the primary winding of a transformer and that induced in the secondary winding?*

Yes; the voltage induced in the secondary winding is always in direct proportion to that in the primary winding. If both primary and secondary windings are composed of the same number of turns of wire, the electromotive force induced in the secondary winding will be practically equal to that applied to the primary winding; in this case there will be no change in the voltage or current. If, however, the secondary winding consists of one-half as many turns as are in the primary winding, the secondary electromotive force will be half that of the primary and the transformer will be of the step-down type. If the secondary winding consists of twice as many turns as are in the primary winding, the secondary electromotive force will be twice that of the primary, and the transformer will be of the step-up type. In a step-down transformer of the ratio mentioned, the current in the secondary winding will be twice that in the primary, and in the step-up transformer of the two-to-one ratio mentioned, the current in the secondary will be one-half as great as that in the primary.

400. *Is there any loss in a transformer?*

Yes; there are three causes of loss: the resistance of the windings, the opposition (called "hysteresis") that the iron in the core offers to the rapid changes in the magnetism which the primary current produces in it, and the induction of stray currents in the core in the same way that current is induced in the secondary winding.

401. *What means are taken to keep down the resistance loss in the windings?*

The length of wire per turn is made as small as possible and, as the resistance of the windings is increased by an increase in their temperature, special means are taken to provide ample radiation. Although in some transformers the heat developed is carried off by a natural draft, in many of them the coils and core are immersed in oil, the oil acting as a medium to conduct the heat from the coils to the surrounding case or tank. If there is sufficient space inside the case the oil will circulate of itself, the cooler oil rising as it becomes more and more heated, and the hot oil on the top descending as it becomes cooled by contact with the inside surface of the case. The efficiency of the case for cooling the oil may be made greater by corrugating it, thereby increasing its radiating surface.

In transformers of very large capacity artificial means are provided for cooling the oil. In some of these the heated oil is cooled by coming in contact with pipes through which cold water is circulated. In others, oil is forced through the cooling pipes, and in still others no oil is used, but ventilation and radiation of the heat are secured by forcing air through the coils and core.

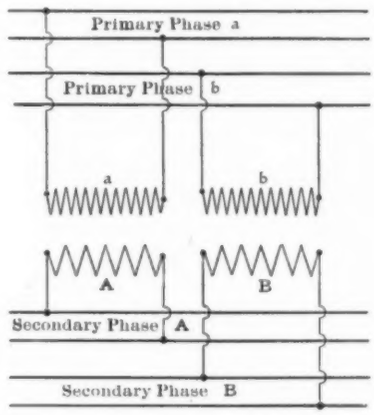


FIG. 85. CONNECTION OF TRANSFORMERS IN A TWO-PHASE FOUR-WIRE SYSTEM

402. *What means are taken to keep down the stray current loss?*

The iron core is built up of very thin sheet iron or steel, and these are insulated from each other by varnish or sheets of tissue paper, and are laid face to face at right angles to the path that the stray core currents tend to follow, so that the currents would have to pass from sheet to sheet, through the insulation.

403. *What means are taken to keep down the hysteresis loss?*

The core sheets are stamped from the

softest iron obtainable, and are worked at a low degree of magnetization.

404. *What means are taken to prevent the high-voltage current in one winding from breaking into the other winding?*

Very high-resistance insulating materials are used between the coils, such as oiled silk, oiled linen, mica, fiber, or micanite. The wires forming the coils are covered individually with several layers

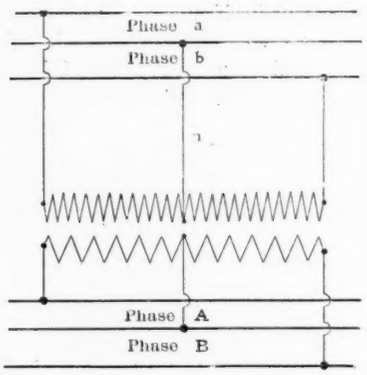


FIG. 86. CONNECTION OF TRANSFORMERS IN A TWO-PHASE THREE-WIRE SYSTEM

of cotton fiber. The oil used in transformers primarily for cooling them is also of advantage in preserving the insulation from oxidation, and in re-insulating the material in case it becomes punctured.

405. *Upon what is the efficiency of a transformer based?*

The efficiency of a transformer may be expressed in terms of its full-load operation or in terms of its all-day operation. The full-load efficiency of a transformer is the ratio between the watts supplied to the primary winding and those delivered by the secondary winding with a non-inductive load. The full-load efficiency curve of a modern type of transformer is shown at *m*, Fig. 84.

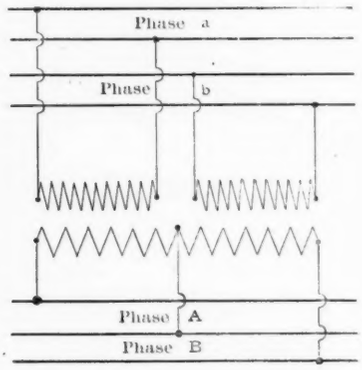


FIG. 87. CONNECTION OF TRANSFORMERS TO OBTAIN A TWO-PHASE THREE-WIRE SYSTEM FROM A TWO-PHASE FOUR-PHASE SYSTEM

The all-day efficiency of a transformer is the average of its efficiencies throughout the range of load supplied by the transformer during 24 hours. An all-day efficiency curve is shown at *n*, Fig. 84.

406. *With a constant primary voltage does the secondary voltage of a transformer remain constant under different load conditions?*

No; as the load on the secondary circuit is increased, the secondary voltage drops. The drop of secondary voltage between no load and full load expressed as a percentage of the secondary voltage at no load is called the regulation of the transformer; its value should not exceed two per cent. for ordinary work.

407. *What causes the drop in secondary voltage as the load increases?*

The resistance of the windings. The greater the current passing through them the greater will be the voltage necessary to force that current through. This voltage is taken from the secondary voltage, leaving less pressure available for useful work outside.

408. *Is a transformer ever wound so that it can be connected to circuits of different voltages?*

Yes; some transformers have two primary windings, which may be connected either in parallel or series so that the same transformer can be used either on a 1000-volt circuit or a 2000-volt circuit. In the former case the primary coils would be connected in parallel, and in the latter case in series.

409. *Are transformers ever connected in series or in parallel with each other?*

Very seldom in series, because there is no difficulty in building them for any voltage now employed, and the first cost is less than if two or more transformers were used in place of one. Transformers are often used in parallel so that each of them may be worked at full load and consequently at highest efficiency.

410. *Must transformers conform to any special requirements in order to work well in parallel?*

Yes; they should be of similar design and size, their percentage of regulation at full load must be the same, and the secondary terminals connected together should have the same instantaneous polarities.

411. *How are the instantaneous polarities of the secondary terminals of transformers ascertained?*

The actual polarities need not be ascertained. By temporarily joining with a fusible wire two of the secondary terminals of the transformers to be connected together, exciting the primary winding and bringing together the two remaining terminals, if the fusible wire melts it proves that the polarities of the terminals joined together were not the same, and that the connections should be reversed; whereas, if the wire does not melt, it shows that the polarities are the same and that the temporary connections may be made permanent.

412. *How many transformers are required in a two-phase four-wire system, and how are they connected?*

Two transformers are required, one in each phase, connected as shown in Fig. 85. Here *a* and *A* are the primary and secondary windings, respectively, of one of the transformers, and *b* and *B*, respectively, the primary and secondary windings of the other transformer.

413. *How many transformers are used for the two-phase three-wire system, and how are they connected?*

Two transformers are used as before, but as shown in Fig. 86 they are connected somewhat differently. The primary windings of the two transformers are connected together and the common conductor *m* of the three-wire system runs to their junction. The secondary windings are connected to the secondary three-wire circuit in a similar manner.

414. *In Fig. 86, what is the relation between the secondary voltage across the two outside wires and that between either outside wire and the center wire?*

The voltage between the two outside wires is 1.414 times that between either outside wire and the center wire.

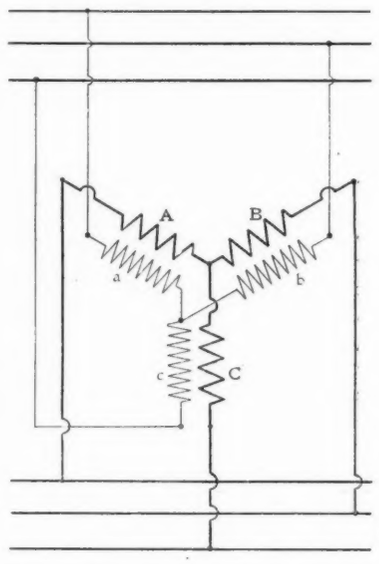


FIG. 88. CONNECTION OF TRANSFORMERS ON A THREE-PHASE STAR SYSTEM

415. *Is it not possible by combining the two forms of transformer connections shown in Figs. 85 and 86 to obtain a two-phase three-wire system from a two-phase four-wire system, or vice versa?*

Yes; such a combination is shown in Fig. 87, where a two-phase three-wire system is shown supplied from a two-phase four-wire system. The reverse of this arrangement is generally used for long-distance transmission of two-phase current, three wires being used in the high-voltage line instead of four, because of the saving of copper.

416. *How many transformers are required for the three-phase star-connected system, and how are they arranged?*

Three transformers are used, connected up as indicated in Fig. 88. Here *a*, *b* and *c* are the primary windings of the three transformers, and *A*, *B* and *C* the corresponding secondary windings. One of

the terminals of each primary winding is brought to a common connection and one of the terminals of each secondary winding is brought to another common connection; the three remaining terminals of the primary and secondary windings are connected respectively to the primary and the secondary circuit wires.

417. *How many transformers are required for the three-phase delta system, and how are they connected?*

Three transformers are used, connected as shown at Fig. 89. As before, *a*, *b* and *c* are the respective primary windings of the three transformers, and *A*, *B* and *C* the respective secondary windings. Their connections to the primary and secondary circuits are obvious.

418. *Has either the star or the delta connection of transformers any material advantage over the other?*

The star connection has an advantage over the delta connection that each primary transformer winding is subjected to only 57.7 per cent. of the circuit voltage. For high-voltage transmission this feature of the star connection allows smaller transformers to be used than with the delta connection, and the insulation has to withstand only 58 per cent. of the voltage; the cost is therefore less.

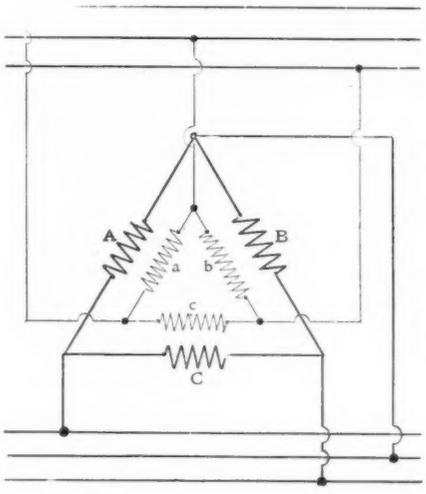


FIG. 89. CONNECTION OF TRANSFORMERS IN A THREE-PHASE DELTA SYSTEM

The delta connection, on the other hand, has an advantage over the star connection where continuity of service is very necessary, because the disabling and removal of any one of the three transformers does not interrupt the three-phase distribution; even if two transformers are removed, reduced power may be transmitted single-phase fashion.

419. *Suppose three transformers, each with a ratio of transformation of* 10 *to* 1, *are connected star-fashion on a three-phase* 1000*-volt circuit, what voltages would be obtained from the secondary* 100 if both primary and secondary wind*-windings if they were connected delta fashion?*

Since the secondary voltage would be

ings were connected up alike, and the ratio of delta to star voltages at terminals is 0.577 to 1, the secondary electromotive force in the case mentioned will be but 57.7 volts.

420. *What secondary voltages would be obtained if the* 10 *to* 1 *transformers were*

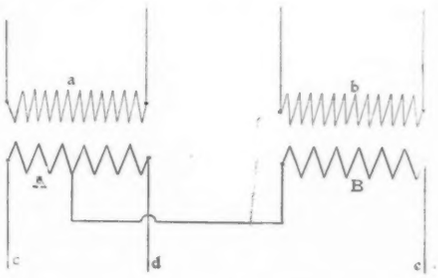


FIG. 90. CONNECTION OF TRANSFORMERS FOR OBTAINING THREE-PHASE CURRENTS FROM A TWO-PHASE SYSTEM

connected with their primaries in delta to a 1000*-volt circuit and their secondaries in star relation?*

There would be induced in each secondary winding $100 \div 0.577 = 173.2$ volts.

421. *Can transformers be connected so as to deliver three-phase currents from a two-phase supply circuit, or vice versa.*

Yes; if the windings of the transformers be designed with this point in view. Referring to Fig. 90, *a* and *b* represent the primary windings of two transformers connected to a two-phase four-wire circuit for the purpose of converting it into a three-phase circuit of the same or different voltage. The secondary winding *A* of the one transformer is wound in two equal sections so that each of them gives one-half of the required voltage in the secondary circuit. The secondary winding *B* of the other transformer is wound so as to give 0.866 times the required secondary voltage. At the three secondary terminals *c*, *d* and *e* three-phase currents are delivered, and between any two of the three terminals the desired voltage exists. By simply reversing the conditions in Fig. 90, currents from a three-phase system can readily be converted into two-phase currents and delivered to a four-wire system at any required voltage.

The *Mechanical Engineer*, London, tells of a company in Glasgow which has introduced a producer-gas plant as a subsitute for petrol engines; which, it is claimed, secures a considerable saving. Either coke or charcoal may be employed, and it is stated that a 30-horse-power vehicle can be run for one hour on 19½ pounds of coke and 2 gallons of water. The engine can be started from the cold in five minutes. The plant consists of a producer, fuel hopper, blower, small pump for feed-water, gas cooler, and air-mixing valves and water tanks.

The Barometric Condenser*

THE PRESCOTT CONDENSER

One more example of the simple type without air pump is the condenser the especial features of which were conceived by Charles J. Davidson, chief engineer of power plants for the Milwaukee Electric Railway and Light Company, and worked out by him in collaboration with

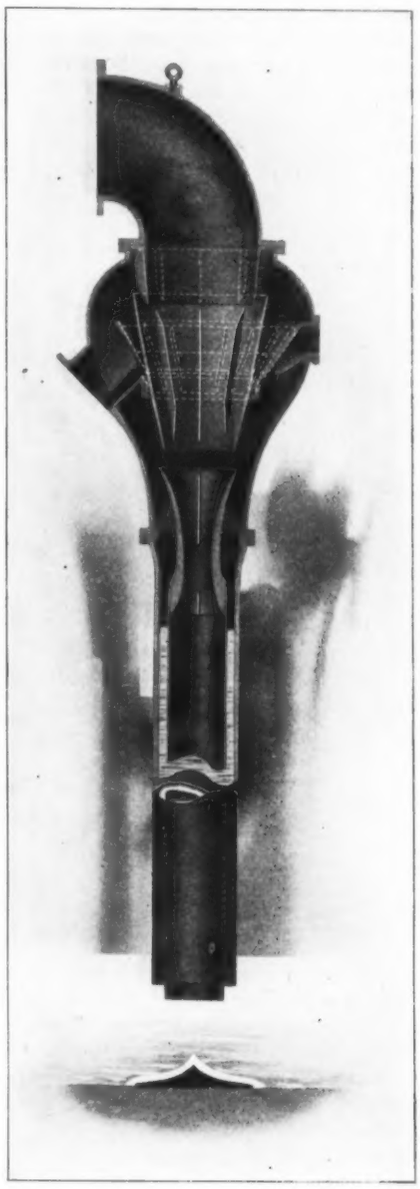


FIG. 12. THE PRESCOTT CONDENSER

the engineers of the Fred M. Prescott Steam Pump Company, of Milwaukee, who build and install it. This condenser, as shown in section in Fig. 12, is divided into two chambers, into the upper of which the injection water enters by the flanged opening at the left, fills the annular space surrounding the funnel and overflows into the latter in a tubular or funnel-formed stream. The exhaust steam entering at the top is discharged downwardly with considerable velocity into this

*Continued from page 5, January.

hollow cone of water, to which it imparts its energy. The interior of the cone is provided with vanes to prevent a rotary or swirling motion of the jet which is narrowed and discharged into the contracted throat below, where it is further contracted, with consequent compression of the air which it contains, and discharged into the central tail pipe. An outer tail pipe takes care of the water from the steam which may condense upon the lower portion of the shell, or any other water which may fail to enter the contracted throat, and is also available for taking care of water in excess of the capacity of the inner tube in the same way as Tomlinson's auxiliary tail pipe. The patent shows, in fact, this pipe arranged at the side of, as well as surrounding, the main leg, and also shows the two pipes joined below the normal water level in the auxiliaries.

The deflector in the bottom of the pipe in the hot-well throws the air to one side, overcoming any tendency to rise through the pipe where the flow is slow. A connection is provided at the right for a dry vacuum pump to draw off any air which may not have been carried off by the jet. The entrance to this pipe is so shielded as to compel the air to pass along the under side of the partition (which is covered above by the incoming cool condensing water) and thus to be cooled and contracted to the smallest attainable volume.

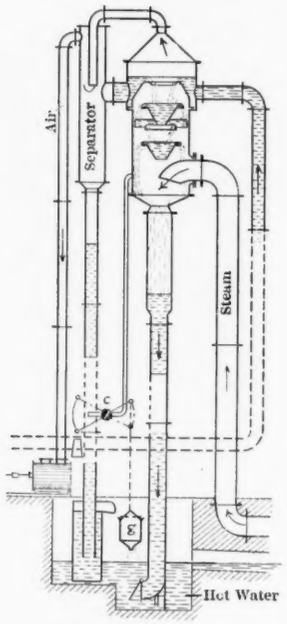


FIG. 13 THE WEISS COUNTER-CURRENT CONDENSER

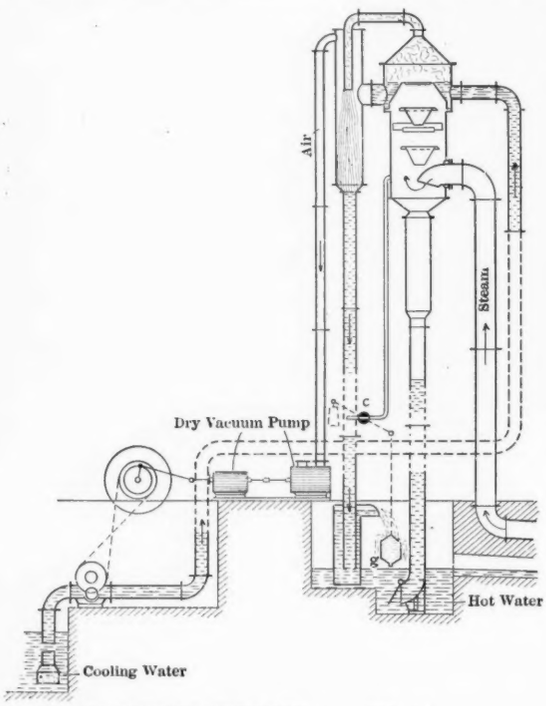


FIG. 14

CONDENSERS WITH DRY VACUUM AIR PUMPS

Franz Joseph Weiss, of Basle, Switzerland, began in the early nineties to develop the possibilities of the barometric condenser for high vacuua. He called attention forcibly to the fact, which has been already pointed out, that the absolute pressure existing in the condenser was the *sum* of the vapor pressure (i.e., the pressure of steam as given by the steam tables for the temperature of the water content of the condenser) and of the air content. The absolute pressure could not get down to that due to the temperature, except in the absence of air, but the lowest pressure in the condenser would be at the coolest point, for here the vapor pressure would be smallest and to this point the air would tend to flow. His condenser was therefore made on the counter-flow principle, as shown in Fig. 13.

The steam, instead of entering at the top as in the types that have been previously considered, enters at the bottom, the exhaust pipe being turned downward to direct the steam into the barometric leg. The injection water enters at the top, as shown, and trickles down over a series of shelves or bowls, while the dry air pump draws from the top. The point of least pressure will obviously be at the top, and to this point any air or vapor not condensed at the entry or carried down into the tube will make its way, encountering cooler and cooler water as it arises, until at last only the air or the absolutely non-condensable vapor reaches the air pump, and this at the temperature of the incoming coolest water. The air is drawn off through a separator which has a barometric tube to discharge the water which comes over with the air. With this arrangement it is possible to get the vacuum, or the absolute pressure, down to that due to the temperature, but as this "ideal point," as the inventor terms it, is reached, the water is apt to rise and pass out by the air pipe at the top, as shown in Fig. 14. In other words, the condenser reverses itself and becomes a parallel-flow instead of a counter-flow apparatus, and having once established itself in this way it will continue to run so until the condenser is stopped and started over again right. In order to do this automatically, a pail *g* is suspended so as to receive the flow from the barometric tube of the separator. This pail has an outlet in its bottom sufficiently large to discharge the normal flow as fast as it is received, but when the condenser reverses, and the water comes in large quantities the pail fills, overcomes the weight at the other end of the lever, and turns the cock *c* connecting the chamber into which the incoming steam is discharged with the barometric leg of the separator, which equalizes the pressures and reverses the action, turning the condenser back into a counter-current.

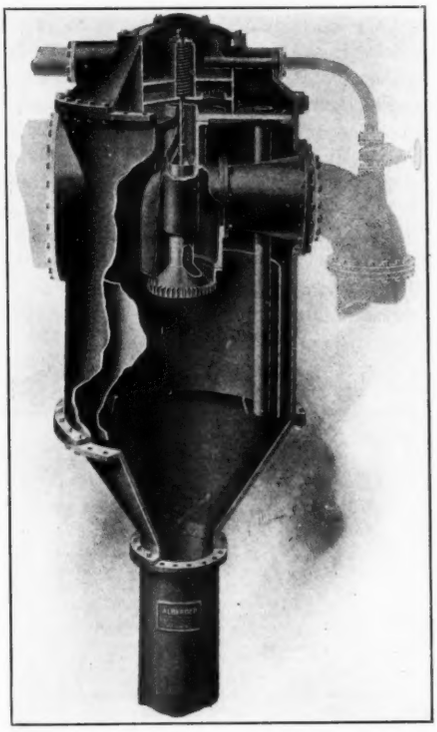


FIG. 15. ONE TYPE OF ALBERGER CONDENSER

Particular stress is put by Weiss upon the fact that the advantage of the counter-current condenser lies not in the reduction of the volume of the air to be pumped by securing the greatest possible reduction of temperature, although that is an advantage, but in the fact that by maintaining the point of lowest temperature at the intake of the dry air pump he establishes a definite and positive flow of air to that point, so that the pump draws air, not steam vapor, and all the air is taken out.

THE ALBERGER CONDENSER

shown in Fig. 15 embodies the essential feature of the counter-current type in chilling the air to the lowest point at the intake of the dry vacuum pump and makes even less attempt to direct the air which enters with the exhaust into the tail pipe than does the Weiss. The condensing water enters at the right and passing downward is finely divided by the spray-cone and brought into intimate contact with the exhaust steam which enters at the

opposite side and fills the entire interior of the condenser. The spray-cone is hung upon a long spring, and by this means adjusts itself automatically to the quantity of water that is entering the condenser, so that when a small amount of water is passing under a light load it will be just as effectually sprayed as when the flow is heavier. The water and such of the steam as it has condensed fall to the bottom of the chamber, where it is brought into contact with that portion of the steam which has found its way through the annular chamber between the outer and the inner shells.

Although no attempt is made, as in the first two cases described, to induce an ejector action by means of the contracted tube, much of the air will be entrained by and pass out with the water. That which does not, passes up through the center of the spray-cone and through the openings with serrated edges to the outlet to the dry air pump. A small portion of the cold injection water is diverted by the pipe shown to the upper chamber, and passing down in an opposite direction to the current of air cools it, reducing the volume which the pump has to handle, and consequently the work which the pump has to do. A large space around the cooler permits of the thorough separation of the water from the air before it goes to the pump, and pipes, one of which is shown, connect this space with the lower portion of the condenser. In case of the failure of the air pump the condensor will still maintain a considerable degree of vacuum.

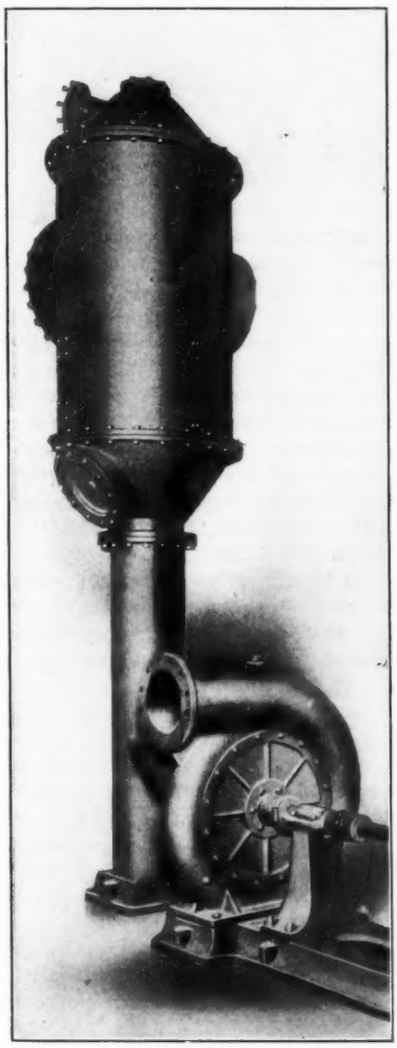


FIG. 16. ALBERGER CENTRIFUGAL CONDENSER

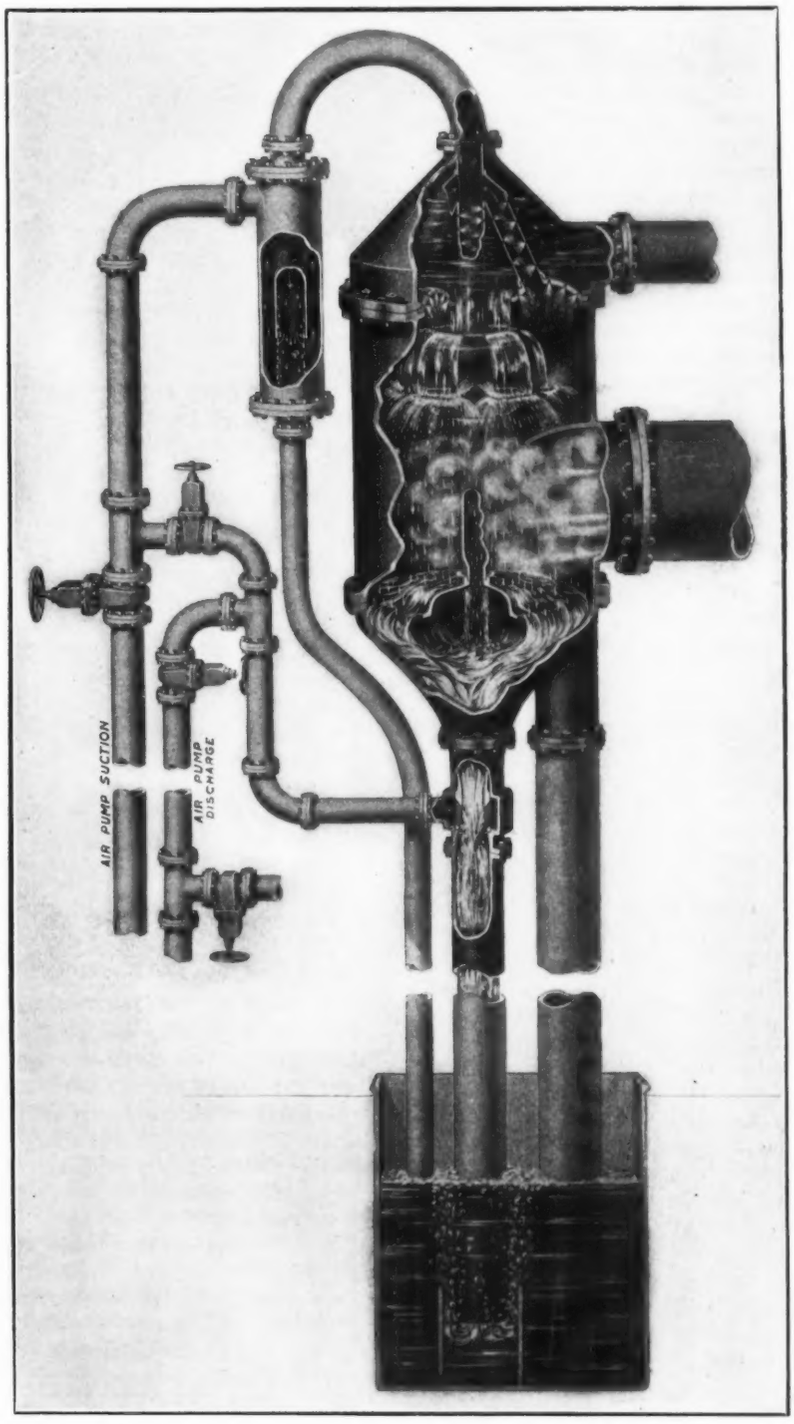


FIG. 17. THE TOMLINSON TYPE B CONDENSER

It was explained in the early part of the article that where advisable the barometric tube could be replaced by a centrifugal pump. Fig. 16 shows a condenser so arranged, as built by the Alberger Company. In steam-turbine practice, with the large ratios of expansion, the exhaust mains become enormous, and it is sometimes advisable to keep the condenser down by the turbine outlet and replace the barometric tube by the centrifugal pump, as shown, rather than to put in a long exhaust main five or six feet in diameter, occupying valuable space, requiring massive supports, expensive to install, and subject to leakage at every pore and joint.

THE TOMLINSON TYPE B CONDENSER

The adaptation of the Tomlinson condenser to the use of the dry vacuum pump

is shown in Fig. 16. This again is of the counter-current type. The injection water enters at the top and is delivered through a central aperture upon the distributor and spray-plate, as shown, becoming finely divided to meet the incoming steam and flowing with the condensed steam over the edges of the umbrella to the tail pipe below. The spray-plate has no openings which are liable to clog. The air which enters with the injector water is separated in the upper chamber at the lowest temperature, i.e., that of the incoming water, and does not come into contact with the steam at all. Much of the air entering with the steam is swept into the tail pipe with the water and discharged as in the types already considered. That which does not so pass off is drawn through the air cooler by the dry air pump, passing through the pipes surrounded by the entering water, cooled to the lowest possible temperature and contracted to the smallest possible volume. To this end the air pipes are not only surrounded by water, but are perforated to let the incoming water mingle directly with the air, and a separator is interposed between the condenser and the air pump to eliminate any water which may be carried out of the condenser. The separator discharges directly to the hotwell through a barometric tube, as in the Weiss.

The air pump, instead of discharging into the atmosphere, is made to deliver into the tail pipe where the vacuum is still sufficient to support the column of water below the point of delivery. The effect produced is that of a two-stage air pump, the tail pipe becoming the second stage. As has been shown above, the delivery of this air in this way demands extra water in the central tail pipe, but at the rate at which this pipe is flowing at anything near normal load, this demand is within the water supplied to meet other requirements. The engraving shows plainly how the air is introduced and directed downward by inclined openings from a chamber surrounding the pipe. The pump is arranged to discharge into the atmosphere in starting up, and to be switched into the tail pipe by the manipulation of the valves shown, or the pump may be by-passed in times of emergency and the tail pipe made to draw upon the upper part of the condenser direct. The auxiliary or overflow tail pipe is also used with this type. The condenser is built up of a succession of flanged sections, all bolted on the outside, with no interior fastenings, and the standard drilling of the flanges allows it to be turned in any direction.

(To be concluded in our next)

The American Society of Mechanical Engineers is now located in the Engineering Building, 12 West 39th street, New York.

DESIGN, CONSTRUCTION AND APPLICATION OF LARGE GAS ENGINES IN EUROPE—XVI

Rules for Testing Gas Producers and Engines

BY FRANZ ERICH JUNGE

The preparation of the following rules for making gas-engine and producer tests was undertaken by a committee appointed from the Verein deutscher Ingenieure, in collaboration with the German Society of Engine Builders, with a view to establishing definite general regulations governing such tests. It is desirable, by specifying the important proportions of the examined plants and the conditions under which the results were attained, to take care that these results are not only applicable to a single case, but that they have general value. To attain this end it is necessary that all data should be given uniformly according to a code of regulations such as those here presented.

The execution of such tests is to be intrusted only to persons possessing the required expert knowledge and practical experience. These persons must make a trial plan, or schedule, appropriate to the individual case in hand which, in many instances, will not require that all of the investigations stipulated in the general code are actually carried out. They must further examine the instruments for measuring or recording purposes as to their fitness and must compile the results. The following rules, the adoption or selection of which must be left to the soundness of judgment of the investigator, are to serve as a basis on which to proceed.

GENERAL REGULATIONS

OBJECT OF INVESTIGATION

(1) The object of a test made on a producer-gas plant can be to determine:

(a) The quantity, composition and calorific value of the fuel consumed;

(b) The quantity, composition and heat value of the gas produced;

(c) The degree of efficiency of the producer-gas plant;

(d) The separate heat losses in the plant;

(e) The quantity of impurities contained in one cubic meter, or one cubic foot of gas (dust, tar, sulphur, etc.,);

(f) The moisture contents of the gas;

(g) The water consumption of the producer-gas plant, either total or in the separate parts;

(h) The mechanical work required for operating the plant, including cleaning apparatus;

(i) The duration or time required for starting;

(k) The stand-by losses during intervals of shutting down at day or night times.

(2) The subject of a test made on an internal-combustion (gas) engine can be to determine:

(a) The indicated capacity and the effective output;

(b) The mechanical efficiency;

(c) The fuel consumption and the heat consumption per horse-power-hour;

(d) The consumption of lubricants, separately for cylinder and engine;

(e) The consumption of water and the heat conducted to the cooling water;

(f) The fluctuations in number of revolutions;

(g) The consumption of exhaust gases.

NUMBER AND DURATION OF TESTS; ADMISSIBLE FLUCTUATIONS

(3) The number and duration of trials are determined by the purpose of the test as well as by considerations of the conditions of installation and operation and must be settled and previously arranged according to paragraphs 4 to 8. For trials of special importance the results of which are decisive for contract tests, for penalties or for premiums, this item is to be treated also according to the significance of the interests connected therewith.

(4) Delivery tests should be made if possible immediately after a plant has been put into actual operation; the delivering firm, however, must be granted a reasonable time for making preliminary trials of their own and for carrying out alterations or improvements then necessary. The duration of this term and other conditions are best agreed upon when making up the delivery contract.

(5) In order to be able to get acquainted with the operation of the plant that is to be tested, and to find time for examining the testing devices employed, and to direct the observers and assistants, it is essential that preliminary trials be allowed.

(6) If the fuel consumption in gas producers is to be determined, the trial run must be extended over at least eight hours in the condition of constancy and without interruptions.

(7) For determining the consumption of liquid or gaseous fuel and provided the condition of constancy is attained, it is sufficient for the higher loads to extend measurements over an hour or so, while for finding the consumption at the lower loads, measurements of even shorter duration are sufficient. To ascertain the condition of constancy the temperature of

the outflowing cooling water must be read from time to time. The previous remarks as to the duration of tests are made with the provision that no interruption or disturbance of the trial takes place, and that intermediate readings show only slightly diverging values for the consumption.

(8) If only the mechanical efficiency of an engine is to be determined, trials of short duration in the condition of stability are sufficient; but at least 10 sets of indicator cards are to be taken.

(9) For researches of special importance at least two tests are to be made, one after the other. They can be considered correct only if no interruptions occurred and if the results show no greater deviations than what can be ascribed to unavoidable errors of observation. The mean of the two results is to be taken as the definite result.

(10) The extent of the difference between the output and consumption that are guaranteed and the results that are recorded which may exist without justifying a claim of failure is to be agreed upon before making the tests (either when making the delivery contract or when preparing the plan or schedule for the trial). When no other agreement has been previously arrived at, the guarantee is regarded as fulfilled if the figure obtained in the test is not more than 5 per cent. below the value on which the guarantee was based. This margin, however, is allowable only for the maximum output which was promised beyond the guaranteed continuous output. The latter must be rendered by the engine under all circumstances. Within the same limits the guaranteed consumption of fuel or water must not be exceeded even when the load during the test is fluctuating, provided that the engine load did not, during the whole length of trial, differ by more than ± 5 per cent., and for a single rest not more than ± 15 per cent. from the condition on which the guaranteed fuel and water consumption were based.

Since it is often impossible when making tests to have the internal-combustion engine work at exactly the effective (horse-power) capacity on which the guarantee agreed upon in the contract is based, it is recommended that the agreement shall specify the expected fuel consumption for the higher and lower outputs. The same provision is preferably made with gas producers.

UNITS OF MEASUREMENT AND DESIGNATIONS

(11) When giving pressure data it must be stated whether absolute pressures or gage pressures above or below the atmospheric are meant. Absolute pressure equals atmospheric pressure ± gage pressure. Atmospheric pressure (zero gage pressure) equals 1 kilogram per square centimeter. (One metric atmosphere = 14.223 pounds per square inch.)

(12) All temperature and heat measurements refer to the thermometer of Celsius, or Centigrade scale.

(13) The mechanical equivalent of heat is taken at 427 meter kilograms (mkg.) = 1 (metric) heat unit = 1 (large) calorie = 3087.77 foot-pounds.

(14) The calorific value of a fuel is to be taken as its lower heating value; that is, the heat which is liberated through the complete combustion of the fuel when the burnt products are cooled down to the original (room) temperature at constant pressure, it being assumed that the combustion water and the moisture contained in the fuel remain vaporized. The calorific value must be based on the unit quantity or weight of original fuel, without deducting ash, moisture, etc., and is to be expressed in metric heat units (1 metric heat unit = 1 large calorie = 3.968 B.t.u.). For both solid and liquid fuels the unit of weight is the kilogram.

The heat value of gaseous fuels is based on 1 cubic meter at 0 degree Centigrade, and 760 millimeters barometer pressure, and must be expressed in calories as "effective" heat value, that is, reduced to 1 cubic meter of actual gas analyzed. If nothing special is mentioned, then it is always understood that the heat value recorded has been reduced to 0 Centigrade and 760 millimeters barometer pressure.

(In this country the general standard so far recommended seems to indicate for "standard gas" a temperature of 60 degrees Fahrenheit, and a pressure of 14.7 pounds per square inch, corresponding to the usual atmospheric pressure.)

(15) The efficiency of a gas-producer plant is the ratio of the heat contained in the gas as produced to the heat of combustion of the total weight of fuel consumed in the plant, both items being computed from the lower heating value. In producer-gas plants having a separately fired steam boiler, it is advisable also to determine the ratio of the heat which is chemically bound in the producer gas to the heat equivalent of that portion of the fuel which is consumed in the producer proper for making such gas.

(16) The unit of measurement used for the power or work output of an internal-combustion engine is the (metric) horse-power, equal to 75 meter-kilograms per second. (One metric horse-power = 0.986 English horse-power.) It must be clearly stated whether the indicated power, or the useful or available power, is meant. If not otherwise designated it is understood that the figures refer to the useful or available output.

(17) The indicated power of the engine or the indicated work is the difference between the total power developed or work done and the indicated power, or work which is consumed within the engine; in short, the difference between the positive and the negative indicated power or work.

The power required at "no load" is the power indicated when no useful work is rendered by the engine.

(18) Mechanical efficiency is the ratio of the useful power to the indicated power of the engine.

(19) All consumption figures are to relate to the hour, and if they shall be compared with the output of the engine they must be based on one horse-power-hour. If not otherwise agreed upon, these data refer to the useful or available output at full load.

EXECUTION OF TESTS

(20) If the quantity of gas made in a producer or the weight of fuel consumed in an engine is to be measured, then all channels or ducts which are not used in the test must be cut off from the piping which leads to the producer and engine that are to be tested. This is best done by means of blind flanges. The active ducts, pipes, gas holders, etc., must be examined with regard to leakage and made tight if necessary. Unavoidable losses due to leakage must be determined, especially with gas channels laid in brick-work.

FUEL CONSUMPTION OF A GAS-PRODUCER PLANT

(21) The kind, number and duration of tests must be agreed upon according to the general rules designated under paragraphs 1 to 10.

(22) The constructive features and the operative conditions of gas-producer plants must be illustrated and explained in the report by drawings, so far as this is necessary for arriving at a sound judgment of the manner of working and of the results obtained.

(23) Before making the test the plant is to be examined as to whether or not it is in good working order.

(24) The quantity of fuel consumed in the gas producer is determined by measuring the weight of the fuel which is charged into the producer during the trials in order that the producer may contain at the end of the test exactly the same amount of heat—either liberated, or chemically bound in the fuel—that it contained when starting the test. To meet this requirement it is not sufficient that the depth of the fuel bed be the same at the end that it was at the beginning; it must also be taken into consideration what influence the ash and the slag left in the producer, the location of the incandescent zone, the formation of fissures and cavities, the closeness or density of the producer charge, and the chemical composition of the burning fuel particles exercise on the heat contents of the producer.

In order to comply with these requirements the following stipulations are to be met:

(25) When starting the test the plant should be in the condition of stability or normal working condition, if possible. This means that after a period of shut-down for cleaning or repairs it should have

been in active operation for one or more days, running on fuel of the same characteristics and size, with the same depth of fuel bed, the same skill of attendance as regards the charging or feeding of fresh fuel and the removing of slag, and under the same load conditions that obtain during the test.

(26) During the trial the producer shall be charged and poked as nearly in accordance with the requirements for attendance as possible. The level of fuel charge must be the same at the beginning and at the end of tests and should be kept constant during the trial. About half an hour before starting and before stopping a test, the slag and ashes are to be removed. If it is impossible to rake out the ashes during the operation of the producer, the plant must be shut down immediately after stopping the test, the slag must be taken out at once and the producer refitted up to the same level that existed when starting the test. The weight of fuel used for this purpose must be added to the consumption.

(27) The fuel consumed during the trial must be weighed, also the fuel which has not been burnt and remains useful; that is, that portion which drops down from above the grate while raking out the slag, and that which is culled out from the ashes as unburnt. The weight of the former may be deducted from the consumption, but not the amount which is taken out from the ashes, nor the coal dust which accumulates in the scrubbers and in the flues between the producer and the engine.

(28) To be able to determine the quantity of ash and slag produced during the trial, the ash box must be emptied before the test. If this is not possible, as when an inclined grate is used, the refuse in the ash box must be equalized before and after the run.

(29) The stand-by losses during intervals of shutting down at day and night must be determined.

(30) In order to get a representative sample of the solid fuel, the following course may be pursued: Of every carload, basket, or other measure of the fuel, put a shovelful in a covered receptacle. Immediately after the test is over, the contents of the receptacle are to be broken, mixed, spread and quartered by drawing the two diagonals of a square. The two opposite quarters are to be rejected, the two others broken up finer, mixed and quartered, and the two opposite quarters rejected. This is continued until a sample of some 5 to 10 kilograms remains, which is preserved, in well closed receptacles, for analysis. In addition to this a number of other samples must be put away in air-tight receptacles for use in determining the contents of moisture in the fuel.

(31) The composition of the fuel shall be determined by elementary analysis. Its contents in carbon (C), hydrogen (H), oxygen (O), sulphur (S), ash (A), and water (W) must be given in percentage of weight referred to the original fuel. The contents, in the fuel, of nitrogen (N) can be disregarded. The behavior of the fuel when being heated is to be determined by a coking test.

(32) The calorific value of the fuel must be determined by calorimetric analysis. An approximate determination of the heat value can be made on the basis of the chemical analysis by employing the so called "association formula:"

$$81\,C + 290\left(H - \frac{O}{8}\right) + 25\,S - 6\,W.$$

TESTING AN INTERNAL-COMBUSTION ENGINE

(33) Kind, number and duration of trials are to be agreed upon according to the general regulations Nos. 1 to 8.

(34) The constructive features and operative conditions of the engine must be so illustrated in the report as to enable one to form a correct idea of the manner of working and of the results of operation. Especially important are the type and capacity of engine, diameter of cylinder and piston-rods, piston stroke, contents of clearance space, and other essential dimensions; the normal rate of revolution and the admissible fluctuations; kind and heat value of fuel for which the engine is intended. The diameter of the cylinder and piston displacement are to be actually measured if this is possible.

The contents of the compression space are preferably determined by filling with water. If it is impossible to state the cubical contents of the compression space then the compression pressure at full load should be given at least. This is done by taking an indicator card while the ignition is interrupted.

(35) Before making the test the engine must be examined internally and externally as to whether or not it is in good working order.

(36) The number of revolutions of the engine is to be determined by a continuous speed counter, the records of which must be noted at certain intervals, and must be checked or corrected from time to time by direct readings. If the velocity conditions of the engine are to be investigated it is essential to determine the following items:

(a) The number of revolutions during the condition of constancy at maximum load and at no load;

(b) The fluctuations in speed at constant load;

(c) The deviation of the rate of revolution from the condition of constancy when the load is increased or decreased according to prescription. These determinations can be executed with apparatus of the character of the Horn tachograph. The fluctuations of speed during the performance of one engine cycle above and below the mean value, expressed in parts of the latter, are to be determined by calculation unless otherwise provided.

The degree of irregularity of the flywheel velocity =

$$\frac{n\ max. - n\ min.}{\frac{n\ max. + n\ min.}{2}}.$$

(37) The useful output can be determined either by brake test or by electrical measurement. The dimensions and weight of the brake should be determined before the trial. The electrical measurements can be made on a generator directly coupled to the gas engine. The useful work is computed from the output rendered by the dynamo. The efficiency of the generator is to be determined after one of the methods as laid down in the "forms for valuating and testing electrical machinery and transformers," published by the association of German electrical engineers. If the efficiency is found approximately by measuring the determinable losses, then an adequate amount (say 2 per cent. of the full-load output) must be allowed for losses not accounted for. The apparatus with which the electrical measurements are executed must be calibrated before and possibly after the test. Whether anything besides this gross amount for increased bearing friction and air resistance of the dynamo shall be credited to the gas engine must be determined separately for each individual case.

Whether, in case the useful output can neither be determined by brake test nor by electrical measurements, the code provision for testing steam engines can be admitted as correct for gas engines, namely, to designate the useful output as the difference between the indicated work and the work consumed at no load, cannot be settled at the present state of development, since results of accurate investigations are not yet available.

(38) Indicators must be connected immediately to the combustion chamber without employing long piping with sharp bends, and one indicator must be provided for every combustion chamber. For this purpose each compression chamber must contain an opening for ¾- or 1-inch Whitworth thread. The same holds true for pumping cylinders. The indicators and their springs must be calibrated before and after the test according to the forms established by the Verein deutscher Ingenieure.

(39) During the test, cards should be taken quite frequently from every combustion chamber and from the pump cylinders. The cards are to be designated by numbers, and the time when each card was taken, the scale of springs used and the number of single cards obtained must be recorded on the cards. At least five diagrams should be taken on one card successively. From time to time diagrams indicated with a weak spring should be taken from the combustion chambers. The indicated work at no load is to be determined immediately after closing the main test and while the engine is still warmed up ready for operation. Care must be taken that the no-load cards are

not taken during an acceleration or during a retardation period of the fly-wheel.

ANALYSIS OF THE GAS GENERATED IN A PRODUCER-GAS PLANT OR CONSUMED IN AN INTERNAL-COMBUSTION ENGINE, OR OF THE LIQUID FUEL USED

(40) The samples for the chemical analysis of the gas must be taken during the trial at regular intervals and as frequently as possible. They must be either analyzed on the spot or preserved in glass tubes closed by melting the ends. The analysis is to determine, in per cent. of volume, the contents of the gas in carbon monoxide (CO), carbon dioxide (CO_2), hydrogen (H), marsh gas (CH_4), heavy hydrocarbons and oxygen (O_2). In addition it is recommended to determine the contents of sulphur (in grams per cubic meter). The gas samples are to be taken from the connection between the cleaning apparatus and the engine.

(41) The heat value of the gas must be determined quite frequently by calorimetric analysis, and the burner of the calorimeter should be fed from the gas-admission pipe without interruption. In suction producer plants this can be done by means of a gas pump drawing from the pipe. If conditions should make it necessary that a sample be taken from the pipe while the calorimeter is shut off, such sample to be later transferred to and burned in the calorimeter, then the quantity of gas burned should not be less than 300 liters (10.59 cubic feet), in order that the calorimeter can at first be brought into the condition of stability also as regards the overflowing combustion water, and in order that at least 100 liters (3.53 cubic feet) remain available for two simultaneous analyses. The suction pump, the gas holder and the piping must be made tight with special care when making a calorimetric analysis of suction gas.

(42) The gas meter of the calorimeter in which the heat value of the gas is determined must be calibrated. For determining the temperatures of the calorimeter water, only thermometers with calibration certificates or others compared with such are to be used. The scales must be divided at least into tenths of a degree.

On the basis of the chemical analysis the heating value of gases which do not contain heavy hydrocarbons can be computed from the following formula, if a calorimetric analysis cannot be made:

$$30.5\ (CO) + 25.7\ (H_2) + 85.1\ (CH_4).$$

(43) The quantity of gas produced or consumed is to be measured by means of a gas bell or gas meter. The cross-sectional area of the bell is to be determined by measurement at several places of its circumference. Consumption tests with the gas bell shall not be made while the latter is exposed to the sunshine.

(44) The gas meter must be calibrated and mounted true with a water scale; it must be so filled that the water level corresponds to the normal filling obtained during calibration. Between the gas meter and the engine a pressure regulator or pulsometer must be installed or a large suction space provided so that the water level shows only small pulsations during the pressure fluctuations.

(45) At intervals corresponding to the duration of test the following readings are to be made: Position of gas bell at three places or the records shown by the gas meter; the pressure in the bell or in the gas meter; the temperature of the gas when entering and when leaving the gas bell or the gas meter and before reaching the engine; the barometric pressure.

(46) If the temperature of the gas is different when measuring the consumption from what it was when measuring the heat value, then the computation must also take into account the increase of volume which is due to the moisture contents of the gas at higher temperatures.

(47) The consumption of liquid fuel must be determined either by weight or by measuring its volume. For determining heat value, composition, and specific weight of the fuel one representative sample is sufficient.

(48) When measuring the fuel consumption of internal-combustion engines, the consumption of lubricating oil for the cylinder is to be determined at the same time.

(49) If the consumption at low loads of a double-acting tandem or twin engine is to be determined, it is not allowable to shut off the gas admission at one or more sides of the cylinders by hand, provided that no other arrangements have been previously agreed upon and are mentioned in the report, or that the governor acts automatically in the way described.

The above extract from the code of rules may suffice to give to the student of the gas-power problem a general idea of the care which must be exercised and of the many niceties which must be observed when testing gas producers and engines in order to arrive at just conclusions. The value of the establishment of standard methods for this kind of work cannot be emphasized too often nor too strongly. Every now and then we read of phenomenal efficiencies recorded in pamphlets and bulletins sent out by manufacturers, and, what is the most deplorable, in papers and magazines which profess to appeal to the technical public. Though the expert can at once distinguish from the manner of execution of a test whether or not the results are correct, it is an unfortunate fact that a great many engineers and, of course, the purchasing public are unable to analyze or recognize the fallacy of some assertions which are put forward under a semi-scientific disguise. These people are deceived and afterward disappointed by the performances of the machines when they fail to come up to the guaranteed figures. It is the duty of all earnest workers in this field to protect the buying public from impositions of this character and to keep in mind the well-demonstrated truth that there is nothing which can do more harm to the commercial growth of a technical innovation than misdirected enthusiasm on the part of the manufacturer, and misused confidence on the part of the consumer.

Some Gas-engine Common Sense

Just how the following remarks by H. W. Jones happened to pass the censor of *The Gas Engine* and get into its pages is a good deal of a puzzle. They are obviously based on practical experience, however, and are well worth reprinting; hence their appearance here.

When a man wants to buy a gas engine, he wants to have some idea how much it is going to cost to run it. Does he get the information? Not on your life! One of these test sheets is placed before him, and staring him in the face he reads:

"OUR GUARANTEE"

Then I. H. P., B. H. P., B. T. U., R. P. M., and P. D. Q. His brain reels; he signs the contract. The salesman hopes for C. O. D., he gets an I. O. U., and then has to sue to get his money. Why? Because the test was translated wrong. Now I am speaking of city gas. Conditions differ, circumstances alter cases, and gas of certain value—say, 650 B.t.u., about the standard—may possibly differ sometimes, but "Our Guarantee" remains the same.

"We guarantee our engine to deliver a H. P. H. (horsepower hour) on 20 cubic feet of gas, at full load." Sounds nice, doesn't it? Only 20. And then later on the same book says: "Don't buy a 10 H. P. engine to deliver full 10 H. P. ten hours per day; you overwork the engines." And the 20 cubic feet per H. P. H. fades away like snow. When the gas bill comes in—! ! ! The agent is telephoned for. He comes, and on hearing the complaint he blames it on the gas. Poor gas. Pumping air in it, etc. Mr. Agent proudly shows "Our Guarantee," and the test at the works. The agent says: "Put your engine any place, in a dark corner or damp basement; our engine needs no engineer; just put oil on her and start her up; don't need any special attention; always ready, etc.," and again shows "our test."

We all know that when a factory tests an engine they always want a dark, damp place and they don't have an engineer anywhere near the test hole. What good is a test to a buyer unless the conditions under which the engine is to be run are the same, or as nearly as possible the same, as under which the engine was tested at the factory?

I have had some little experience with gas (city gas) engines in a practical way. I have never yet had one complaint either as to the working of the engine or the

size of the gas bills, where the engine was set in a clean, dry, light place, and where it had proper care and attention. By the way, proper care and attention consists, in addition to other things, of the engineer taking the statement of the gas meter each day at a certain hour. If the consumption is larger than it should be any one day, find the cause—it is there.

Everyone who has had experience with about twenty to thirty different makes of engines knows, and knows it well, that "city gas" [and any other kind—Eds.] must, in order to produce *proper* results, have the ignition set at the *proper* time; must have the *proper* compression in the cylinder: must have the valves open and closed at the *proper* moment; must have *proper* lubrication—and, especially in the cylinder, too little is bad, too much is worse, and yet I've heard and re-heard that gas engines do not need an engineer. This should be stopped and the manufacturer should stop it. Gas engines *do* need engineers. Not steam engineers, not gas engineers—but gas-engine engineers, and the better the engineer the better the results.

"Cleanliness is next to godliness"— the man who invented this saying must have been tinkering around gas engines. Now, I've found fault with this "test question" and with placing engines to work [under conditions] almost exactly opposite from the conditions under which they were tested. I offer a better way—the factories to order their representatives to say: "Our engine is a machine that requires attention and care as to adjustment and *cleanliness*, and when these are given, the results will be pleasing to you, and gratifying to the makers, and that when you do care for your engine you won't need any guarantee, but if you don't take care of your engine you will find that our guarantee is so worded that it won't do you any good, and our test is made under the best conditions, and in order to get the best out of our engine you must treat it kindly, because it was raised a pet." Then, and not until then, will the user of an engine urge his neighbor to go and buy the——gas engine, as it is a great engine and it does exactly as they say it will do.

The new municipal generating station at Johannesburg, which is to supply railways, lighting and power service over an area of nearly one hundred square miles, is now approaching completion. Producer gas will be used in gas engines of 1000 horse-power and 2000 horse-power. There have been some difficulties and a good deal of delay in getting this plant started. The usual starting difficulties of any new station have been accentuated by local conditions, by distance from headquarters, by the loss of a shipload of material at sea, and by the unfortunate death of the contractor's chief engineer.

THE LARGEST STEAM-TURBINE POWER PLANT IN EUROPE

The St. Denis Station of the Societe d'Electricite de Paris

BY FRANZ KOESTER

The largest steam-turbine power plant on the continent of Europe was recently completed in Paris, and it is an interesting specimen of modern power-plant design. It is located in the northern part of Paris on the Seine, and was installed to supply power for part of the street-railway system of the city, and belongs to the Société d'Electricité de Paris, which sells power to the operating railway company. This plant will have an ultimate normal capacity of some 60,000 kilowatts, supplied from four 5000-kilowatt and six 6500-kilowatt turbine-generator units. The turbines are of the Brown-Boveri-Parsons type. The equipment at present installed, however, consists of only four 5000-kilowatt generating units and one 300-kilowatt exciter unit.

The general layout of the power plant is that usually found in modern turbine generating stations, the boiler house being at right angles to the generating room. As the size of the generator room was materially decreased by the use of turbines, as compared with the space requirements of reciprocating engines, it was decided to adopt marine type boilers in order to keep down the size of the boiler-room. The boilers, which are of the Babcock & Wilcox type, are arranged in four rows, with two firing aisles between them, there being a suspended coal bunker over each aisle. The coal is conveyed to these bunkers by traveling buckets, which on their return trip remove the ashes from the basement.

The boilers are of the well-known marine tubular type, with one drum across the front of each boiler setting, 17 feet long and 52 inches in diameter. There are 33 sections of 14 tubes each, and the tubes are 10 feet long and 3¼ inches in diameter. The total heating surface per boiler is 4500 square feet, and the floor space is 267 square feet. Each boiler is equipped with two mechanical chain-grate stokers of 35 square feet area each. A superheater of seamless drawn tubes is provided above the boiler, with a heating surface of 640 square feet.

Directly above each row of boilers, 23 feet above the boiler-room floor level, is a gallery containing economizers. The main steam pipes, 12 inches in diameter, are run in the rear of the boilers, and supply steam at 175 pounds pressure and 680 degrees Fahrenheit.

Between the boiler-room and the turbine-room, and extending the entire width of the former (137 feet) is a room 20 feet wide, containing water tanks, meters, etc., while on the opposite side of the turbine-room, which is 65 feet wide, is a separate building for switchboards, executive offices, measuring instruments, etc. This building is 620 feet long and 25 feet wide.

Fig. 1 is a cross-section of the power house, in which the arrangement of the turbines and their auxiliaries is shown very clearly. Fig. 2 shows the floor plan of the generating room, and Fig. 3 is a side elevation of a turbine unit and its auxiliaries. The turbines are arranged in two rows, with the steam intakes of the two lines turned toward each other. This arrangement was adopted in order to make the live-steam connections as short as possible. The surface condensers, which have cylindrical casings, are placed in the foundations directly below the turbines, and are connected to the turbines by short pipes of 70 inches diameter. In the basement below the main generating room are located the cooling-water and air pumps. These pumps and all other auxiliaries are electrically driven. Fig. 6 shows one of the motor-driven air pumps.

As will be noticed in Fig. 1, the water intake and discharge system is somewhat costly; on account of the water level and grades it was necessary to dig a pit for each condenser 60 feet deep and 6 feet in diameter. The supply tunnels are located at the extreme sides, while the discharge tunnels are located in the center of the foundation work, directly beneath the generating-room floor. Adjacent pits are connected by means of tunnels; each pit is provided with two sluice gates to control the intake from the two supply tunnels. As will be noticed, the centrifugal circulating pumps are located at the extreme sides of the sub-basement, just below the condenser-room floor level, while their driving motors are located on the condenser-room floor.

The equipment of the power house is divided into complete separate units, each 5000-kilowatt turbine having its own surface condenser, circulating-water pump and air pump, as well as its own group of five boilers. All four complete units have one exciter set which has a capacity of 75 per cent. of the entire ultimate plant output. There is no reserve exciter unit, but, in case of a break-down, current is drawn from a storage battery with a capacity of 1300 ampere-hours.

The turbines which develop normally 5000 kilowatts have a maximum capacity of 6000, and run at a speed of 750 revolutions per minute. The guarantee of these turbines operating under 175 pounds steam

pressure at 570 degrees Fahrenheit at the throttle and 27 inches of vacuum is 14.9 pounds per kilowatt-hour, and there is no doubt that under actual operating conditions they will show a still better economy on account of the high and strictly enforced penalty clauses common on the continent of Europe. Directly on the shaft of the turbine is mounted the four-pole field magnet of a three-phase, 25-cycle alternator. The entire over-all length of the turbo-generator unit is about 74 feet. Figs. 4 and 5 show one of the units as viewed from each end.

The turbine is equipped with safety devices, which, if the speed rises beyond 15 per cent. above the normal rate, shut the turbine down. A sudden reduction of the load from full rating to zero will cause the speed to increase not more than 3½ per cent. As already stated, the entire auxiliary outfit is operated by electric motors. The circulating-water pump is driven by a vertical 130-horse-power motor, running at 310 revolutions per minute, and the wet-air pump is driven by a 50-horse-power horizontal motor.

A 300-kilowatt direct-current turbine-generator supplies the exciting current for the alternators, and also furnishes current for lighting the station and driving the motors of the various pumps, crane, etc. The turbine runs at 2750 revolutions per minute, and is direct-connected to a 220-volt generator. The turbo-generator unit is provided with an individual

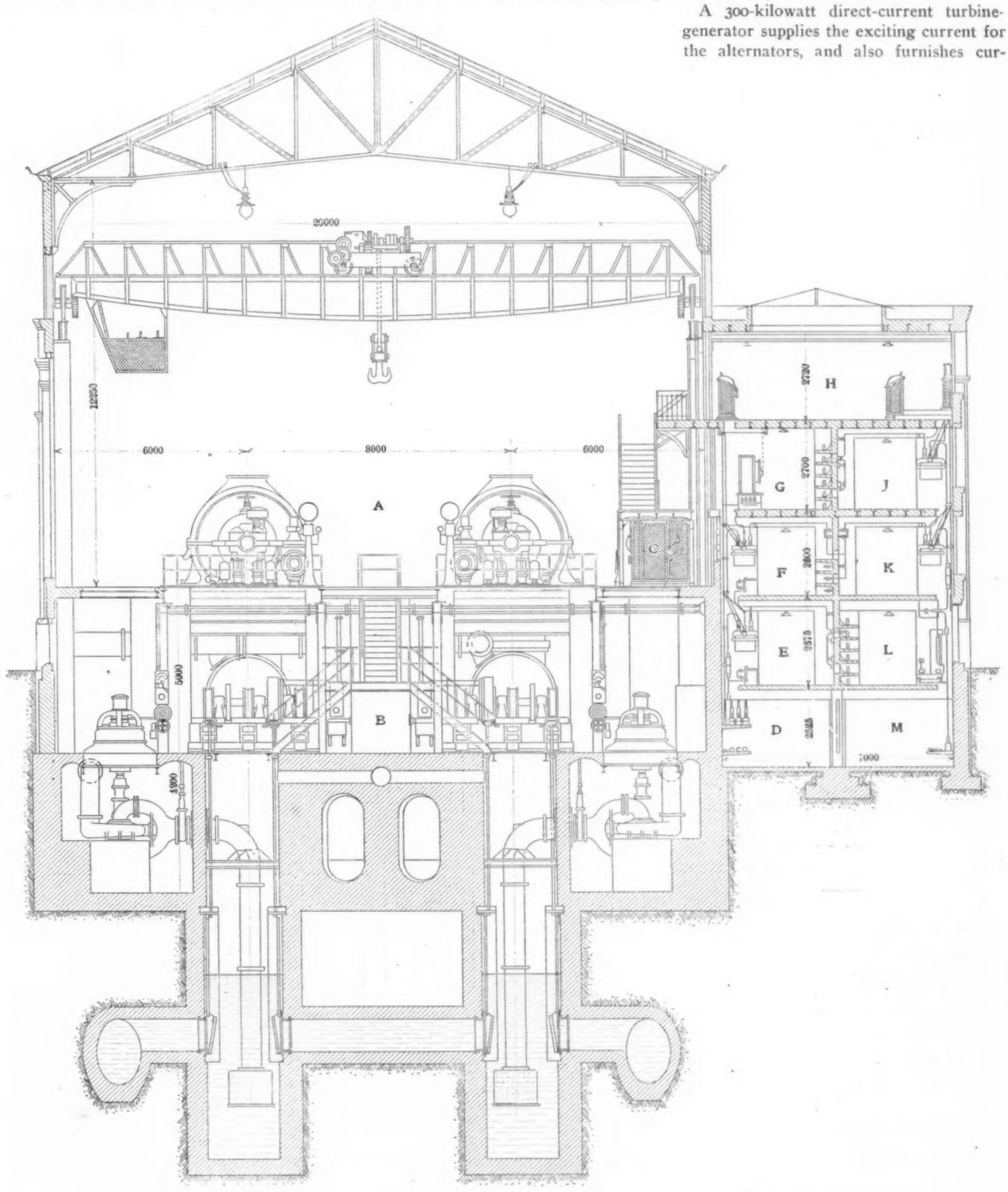


FIG. 1. CROSS-SECTION OF ST. DENIS STEAM-TURBINE POWER PLANT, PARIS, FRANCE

condenser and outfit of pumps, the same as the main units. The centrifugal cooling-water pump is driven by a 16½-horse-power motor at 1520 revolutions per minute, and the vacuum pump is operated by a 9-horse-power motor running at 800 revolutions per minute.

There are also two motor-generator sets, each comprising a 375-kilowatt generator direct-connected to a 540-horse-power synchronous motor, running at 375 revolutions per minute.

The storage battery previously mentioned consists of 126 Tudor elements, with a capacity of 1300 ampere-hours. The booster used in connection with the battery consists of a direct-curent motor direct-connected to a direct-current generator of 600 ampere capacity and a maximum of 120 volts.

The switchboard, as already stated, is located at the side of the generating room, and is equipped with the usual modern instruments and devices for controlling and measuring the current.

All the turbines, as well as the entire switchboard and electrical equipment, were built and installed by the Brown-Boveri Company, of Baden, Switzerland, who will also supply the future installation of three 6500-kilowatt three-phase, and three 6500-kilowatt two-phase alternating-current turbo-generators, now under course of construction. The installation of these additional units, when completed, will give the plant a normal capacity of 60,000 kilowatts, or 80,000 horsepower, and it will then be the largest steam-turbine power plant in existence, if a larger one is not built in the meantime.

It may be of interest to mention the rapidity with which the 5000-kilowatt turbo-generator units were installed. One of the units arrived at the power-house on a Saturday evening and eight working days later it was ready for operation.

Hints for Users of Condensers

The following hints are offered by the condenser department of the Allis-Chalmers Company:

The greatest care is necessary in the design, construction and maintenance to keep the system free from air leakage; for the chief difficulty in maintaining a vacuum arises from such leakage. The leaks are inward, therefore invisible and hard to locate. The designer or the constructor often does not realize all that is here implied, judging from the results sometimes produced.

The application of two or three coats of elastic paint on clean surfaces will elimmate many of the leaks in the exhaust-pipe lines, but there still remain several places for air leaks which are not so easy to remedy. The stuffing-boxes on the rods and exhaust-valve stems of the low-pressure cylinder are apt to be sources of trouble. This, however, can easily be eliminated by using double metallic pack-

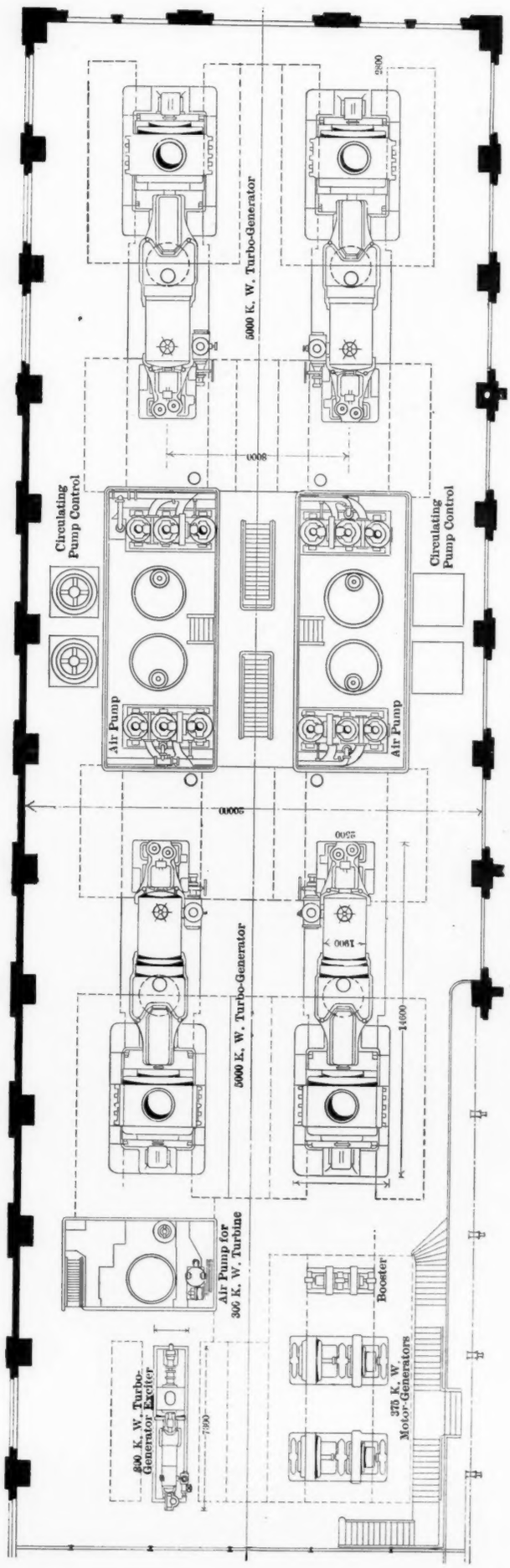


FIG. 2. FLOOR PLAN OF GENERATING ROOM, ST. DENIS POWER PLANT

ing on both rods and valve stems, and piping the steam from the receiver to the space between the two sets of packing, thereby forming a steam seal on the rod. With such a seal it is impossible for air to leak into the system through the stuffing-boxes.

Another source of air leakage into the condensing system is through the water supply. The stuffing-boxes on the pump should always be sealed with water, and all joints on the suction line must be made up tight. The line should be carefully painted before burying it in the ground. The higher the suction lift, the more care should be exercised in this work, because all air drawn in through the suction line which is subject to vacuum is discharged directly into the condenser. The fact that the pipe may be buried several feet underground does not protect it from air leakage.

A further source of air being drawn into the condensing system is with the water through the suction line of the pump. All suction lines should be submerged not less than six feet and more if

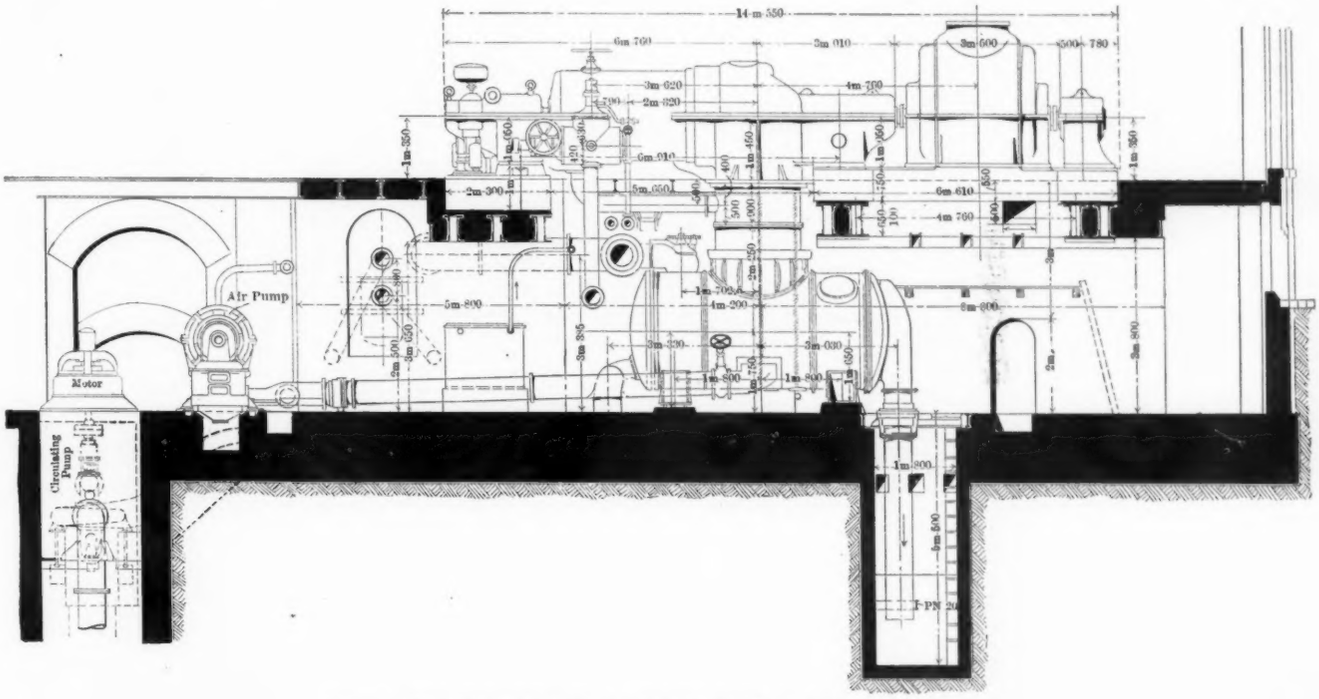


FIG. 3. SIDE ELEVATION OF A TURBINE UNIT, ST. DENIS POWER PLANT

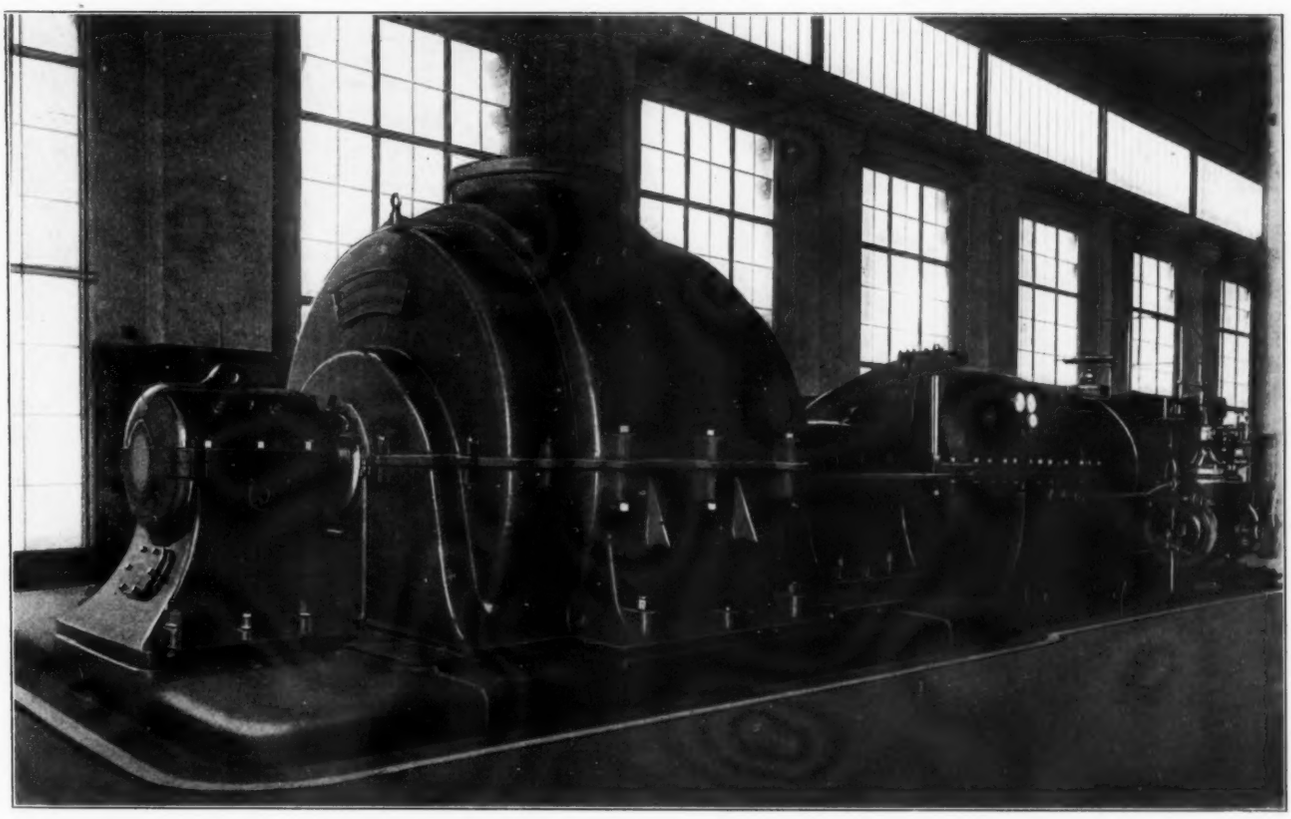


FIG. 4. ONE VIEW OF TURBO-GENERATOR, ST. DENIS POWER PLANT

possible. The end of the suction pipe should be increased to at least twice the diameter of the pipe to reduce the velocity of the water entering the line. In a well 12 feet in diameter, a 12-inch suction line was run to a centrifugal pump. The end of the pipe in the well was submerged 8 feet, but not increased in diameter. It had been observed that dry sawdust, thrown on the surface of the water, was sucked in small eddies down into the suction of the above pump. If sawdust was drawn down, it is clear that air was drawn with it. The amount of air coming down the tail pipe of the barometric condenser, discharging into the hot-well, showed plainly that there was air leakage somewhere in the system, but the source of it was not located until the action of the water in the well was discovered by the engineers in charge of the plant. The remedy for this condition was to enlarge the diameter of the suction pipe in the well.

The Invention of the Steam Jet—An Impossible Saving

W. E. Crane, our sometimes contributor, is very serious-minded when his pen is in his hand. The good stories which he tells from time to time he does not consider of sufficient account to put upon paper. On the occasion of a recent call the conversation turned upon processes of combustion where a forced draft is used.

"I suppose you know," observed Mr. Crane, "how the steam blower came to be invented."

We confessed our ignorance, and the narrator continued: "During the civil war the price of coal ran up to sixteen or twenty dollars per ton, and people were driven to burning yard screenings and anything alse they could get. This required forced draft, and it occurred to some genius that he could put up a fan in front of a steam jet, that the steam jet would blow the fan around and that the fan would pull in air and blow it into the fire. A number of these devices were introduced and worked successfully, but from one cause or another the blades of the fans got knocked off or the rapidly running bearings gave out, and the fans were put out of business; whereupon it

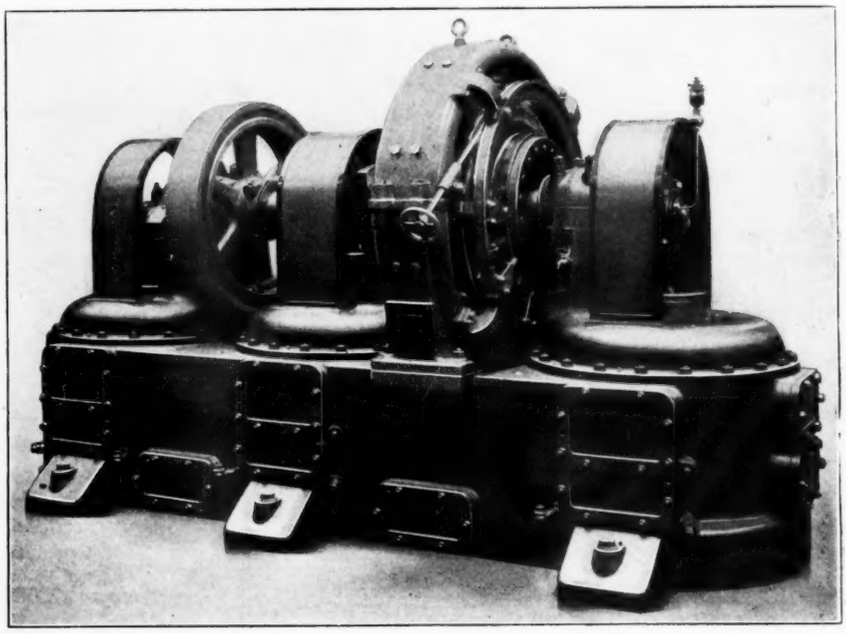


FIG. 6. MAIN TURBO-ALTERNATOR GROUP, ST. DENIS PLANT

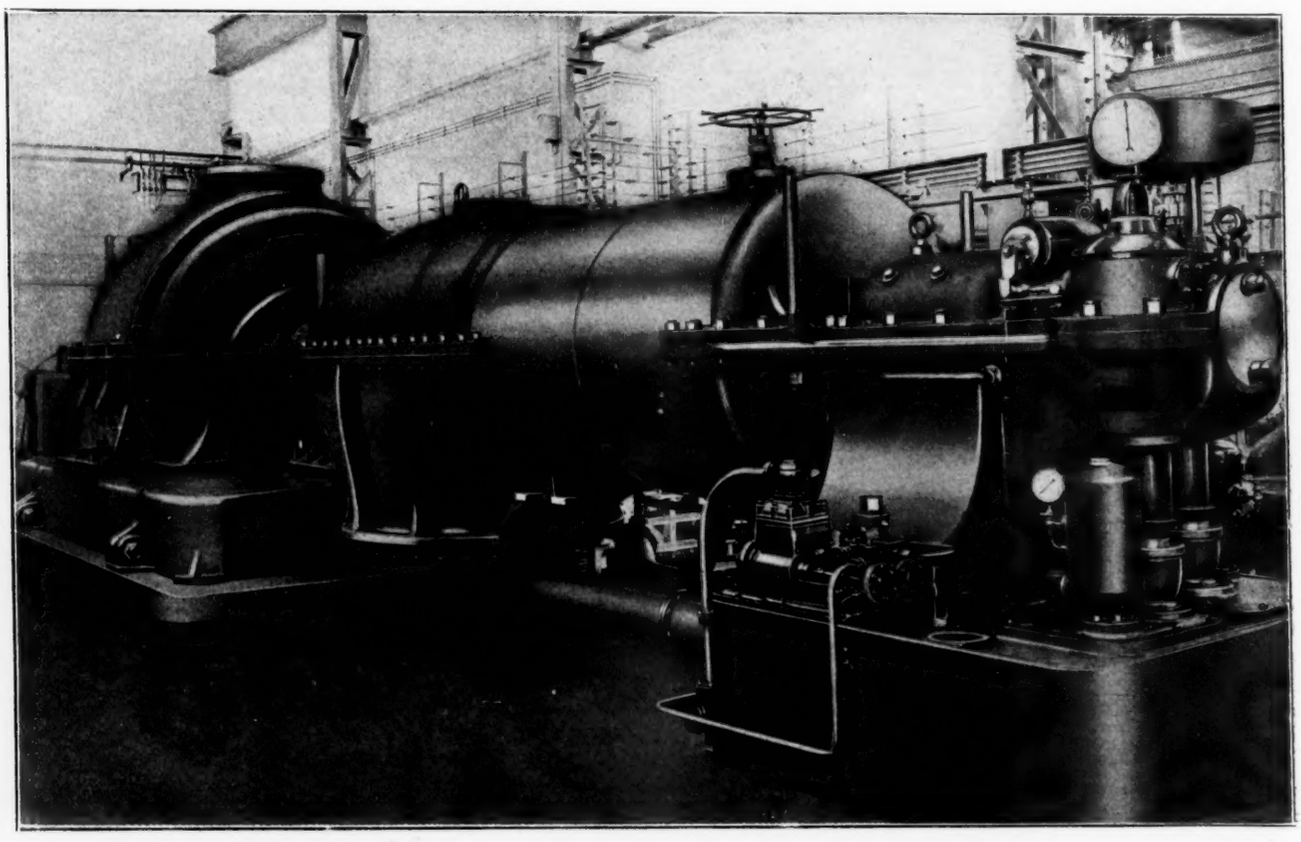


FIG. 5. TURBO-GENERATOR, ST. DENIS PLANT, VIEWED FROM INTAKE END

developed that the steam jet went on pulling in air just the same whether the fan was there or not, and we have had it with us, with more or less modifications of the jets and their distribution, ever since."

During this period the plant where Mr Crane was employed adopted wood as a fuel, and one day he was visited in his engine-room by the promoter of one of the steam-jet blower systems who agreed to guarantee to save the plant five dollars per day by the installation of the blowers. After leading his victim to reiterate this promise a number of times, Mr. Crane informed him that they got wood for four dollars and fifty cents a cord and used three-quarters of a cord a day. The system was not installed.

Shafting

Shafting is subjected to two important strains, to wit: torsional and transverse. Of the two, we believe, from a close observation, that the transverse strain should be guarded against more carefully than is the usual practice, as it is due to lack of proper attention to this point that so many shafts are broken. The torsional strength of shafting, or the resistance to breaking by twisting, is proportional to the cube of the diameters.

The transverse strength, or resistance to bending, is proportional to the fourth power of their diameters and varies inversely in proportion to the load, and also to the cube of the distance from center to center of bearings.

Head shafts carrying main receiving pulleys, rope wheels or gearing, should be enlarged or swelled, so that the bottom of the keyseat in the shaft will not be under the diameter of the body of the shaft.—Geo. V. Cresson Company's *Catalog B.*

Starting Motors

BY W. H. WAKEMAN

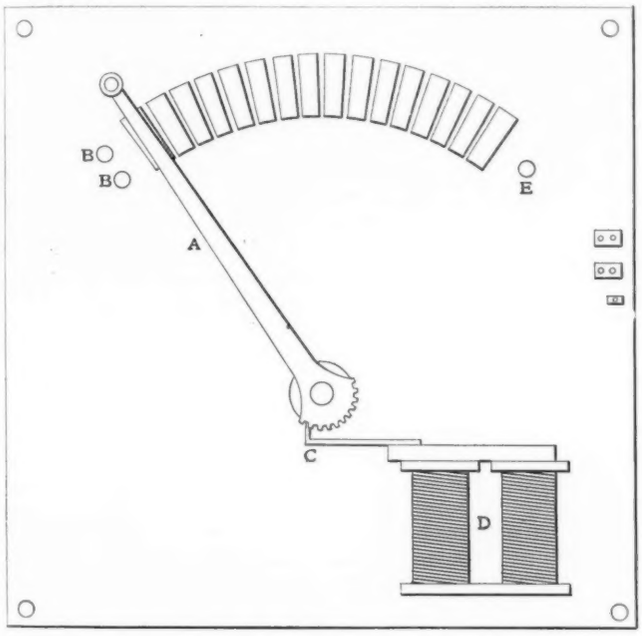
I wish to call attention to the good and bad ways of starting motors that are in use in various plants. Fig. 1 represents the starting lever of a motor controller frequently used in shop and factory practice. Rubber-covered buffers, on which the lever rests when not in use, are shown at *B* and *B*. The lever *A* is shown as moved away from these buffers and resting on the first segment of the rheostat, while the pawl *C* is in the first notch, held there by action of the magnet *D*. As the controlling switch has already been thrown in, bringing the lever into this position allows a few amperes of current to pass, and the motor starts off slowly.

After the motor has run this way for a few seconds, the lever is moved to another segment, as shown in Fig. 2, which cuts out a section of resistance and increases the speed. In a few seconds more it is moved to another segment, and so on, until the desired speed is obtained. This is an ideal way of starting a direct-current stationary motor, and the whole operation can be accomplished in one minute without giving the machine a jerky motion, or bringing any unnecessary strain on the parts.

In contrast to this is the plan (or lack of plan) adopted by careless operators who do not give these matters due attention. The switch is closed and the lever thrown around to the third or fourth segment at once, without regard to any systematic operation of the machinery. This allows such a heavy current to pass that the motor starts off with a jerk and the belt squeaks and slips, which indicates excessive strain on the whole machine, so that the operator waits for perhaps a minute to give the motor a chance to come up to the speed corresponding to the position of the lever, then moves it along two or three segments more, only to repeat the above rough performance, and so on, until full speed is attained.

Motors that are used in this way do not last as long as when the first-mentioned plan is followed, and as it takes fully as long and usually longer to get a motor up to speed, there is no real excuse for the practice, so far as a disinterested observer can discover.

I am much inclined to think that if the men who handle good machinery so carelessly had to keep it in repair at their own expense they would be more careful, at least after paying a few unnecessary bills, but as a rule all they have to do is to report it to the engineer, who must make repairs himself or supervise others who do the work.



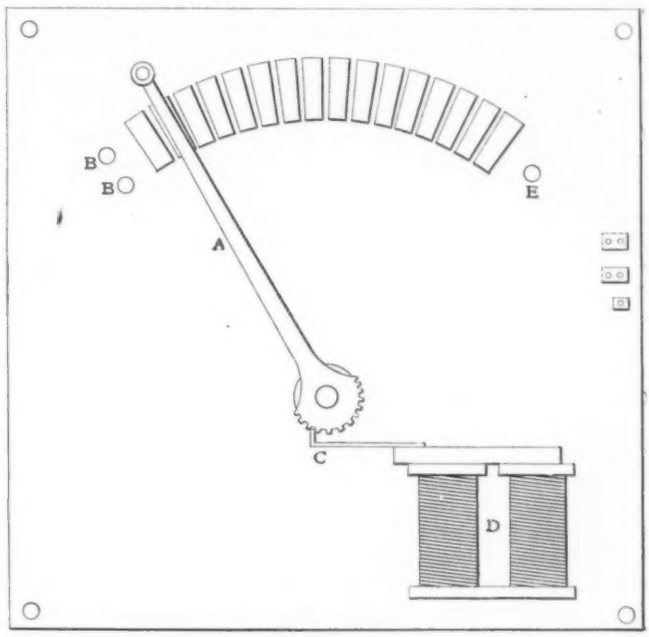


FIG. 1. ILLUSTRATING IDEAL WAY OF STARTING A D.-C. STATIONARY MOTOR. FIG. 2.

Marriage on a Chimney

According to a telegraphic despatch from Peoria, Ill., dated December 22, Carroll Applegate and Miss Alta Gale, of that city, were married on December 21 at the top of the new 200-foot concrete smokestack recently built for the Peoria Gas and Electric Company, and the ceremony was witnessed by thousands of spectators from the ground. Police Justice Fox performed the ceremony, and 12 persons witnessed it on top of the stack. The party was transported to the top in a bucket. The affair was the occasion of the dedication of the stack. One week previously the company had advertised that it would present to any couple married on top of the new stack a gas stove, fixtures for their home, $25 in cash and enough more to pay the minister's fee. Seven couples applied, but Mr. and Mrs. Applegate were first.

THE HYDRAULIC ELEVATOR

Horizontal-cylinder Machines—Comparison of the "Pushing" and "Pulling" Types—Counterbalancing the Hoisting Ropes—General Arrangement of Low-pressure Systems

BY WILLIAM BAXTER, JR.

Horizontal-cylinder elevators are divided into "pushing" and "pulling" machines. A "pushing" machine is one in which the piston pushes the traveling sheaves away from the cylinder, and a "pulling" machine is one in which the piston pulls the traveling sheaves toward the cylinder. In Fig. 16 a pulling machine is illustrated. In this type of elevator, when the car is at the bottom of the well the traveling sheaves are at *D*, and the piston is at the front end of the cylinder *G*. When the car ascends to the top of the building, the traveling sheaves and cables reach the position shown by the broken lines. It will be evident that the pressure water enters the cylinder between the piston and the cylinder-head and forces the piston to the right, and through the piston-rod the traveling sheaves *D* are pulled toward the cylinder. The weight of these sheaves is taken by the roller *E* and guides *F*. The mode of operation of a pushing machine is shown in Fig. 17. Here it will be obvious that the water entering the cylinder between the piston and the cylinder-head forces the piston to the right and, through the rod, the traveling sheaves *D* are pushed away from the cylinder and reach the position indicated by the dotted lines when the elevator car reaches the top of the building.

Which of these two designs is the better, it is difficult to say. The pushing machine obviates the use of a stuffing-box around the piston-rod, but this is offset by the fact that the diameter of the sheaves must be greater than that of the outside of the cylinder, in order that the lifting ropes may clear the latter; this can be readily seen from Fig. 17. In both designs, the stationary sheaves *B* have to be placed at the side of the elevator well, in order that the lifting ropes may run up in the space between the wall of the well and the side of the car. On this account the cylinder of the pushing machine is located nearer the well than that of the other type. In some buildings the pumps, tanks, etc., are located near the elevator well, but in others they are some distance away, so that in some cases it is advantageous to have the cylinder near the elevator, while in others it is not, as it is always desirable to have it as near to the pumping system as possible, to reduce the length of the pipe connections. The pulling machine is not as well adapted to high pressures as the pushing machine, on account of the packing around the piston-rod, and on that account the latter type is the only one that has been used in high-pressure systems.

Horizontal cylinders are not provided with circulating pipes because none is required, both ends of the cylinder being

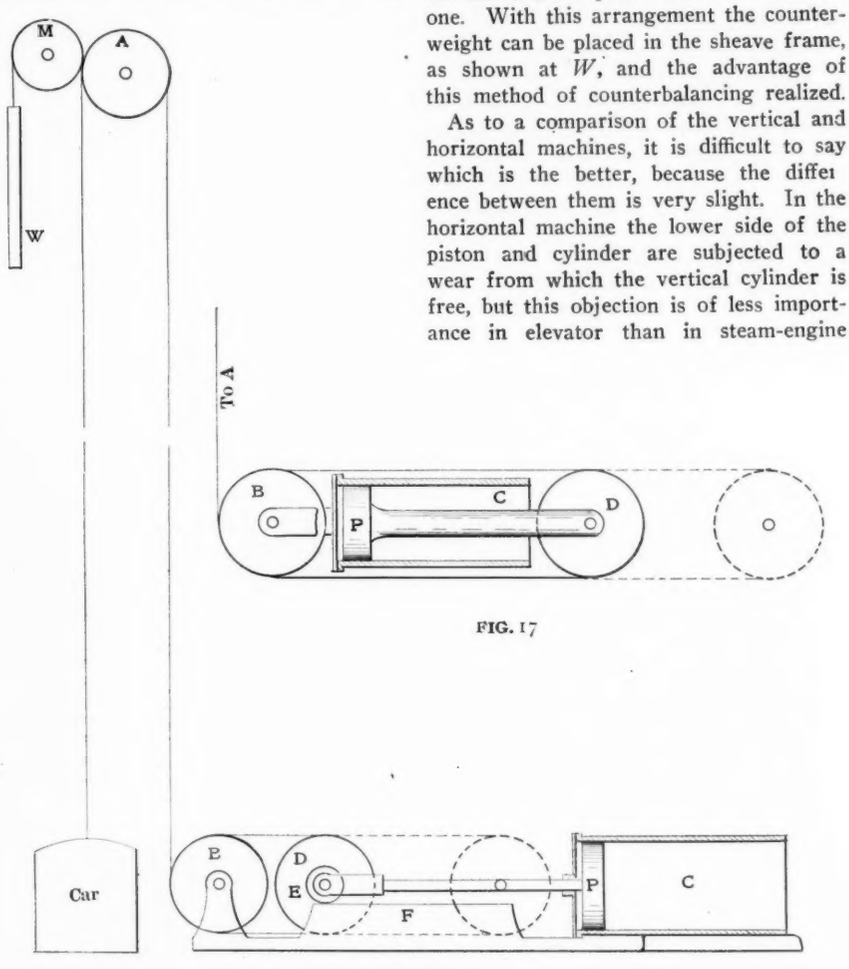


FIG. 17

FIG. 16

on the same level. The pressure water passes in and out of one end of the cylinder only, and no water reaches the other end except what leaks by the piston.

Horizontal-cylinder elevator systems are counterbalanced with an independent weight, as shown at *w* in Fig. 16. As can be readily seen, no counterbalancing can be done in the traveling-sheave frame, because this moves in a horizontal direction. Horizontal machines are geared from six-to-one up to twelve-to-one. The sheaves by means of which the gearing is effected are not located one behind the other, as in the vertical machines, but are placed side by side on axles that are at right angles to the axis of the piston-rod, as indicated in Figs. 16 and 17. The number of lifting ropes used is two, and each one of these passes around a set of sheaves on one side of the center line of the piston-rod, so that the pull of the ropes may not act to twist the sheave axle out of position. When it is desired to obtain a gear higher than twelve-to-one, the general practice is to use a machine of one-half the desired gear, and double this by using a traveling sheave in the elevator well in the manner illustrated in Fig. 18. With this arrangement, if the gear at the cylinder is eight-to-one, the gear between the car and the piston will be sixteen-to-one. With this arrangement the counterweight can be placed in the sheave frame, as shown at *W*, and the advantage of this method of counterbalancing realized.

As to a comparison of the vertical and horizontal machines, it is difficult to say which is the better, because the difference between them is very slight. In the horizontal machine the lower side of the piston and cylinder are subjected to a wear from which the vertical cylinder is free, but this objection is of less importance in elevator than in steam-engine cylinders, as the latter are worn more in a life-time. In the vertical-cylinder in a life-time. In the vertical-cylinder machine the piston friction is greater because the stroke is longer, but to offset this in the horizontal machine the sheave-and-rope friction is greater on account of the higher gear. In practice the type of machine used is in most cases determined by the dimensions of the building. If the building covers a good deal of ground, and the space in the basement is not as valuable as elsewhere, the horizontal machine is used. If the building

is high and stands on a small lot, the vertical type is preferable, and if the floor space is very small, it may be the only type that can be used on account of the length of the horizontal machine being greater than the greatest dimension of the building site.

When horizontal machines are installed in buildings where the space is contracted, it is common practice to stack the machines one on top of the other, making what are called double- and triple-deckers. When vertical cylinders are used, they are sometimes located in the elevator well at one side of the space in which the elevator travels, but this arrangement is not desirable because the noise made by the water passing in and out of the cylinder is objectionable; moreover, if the cylinder springs a-leak, which is not a rare occurrence, the passengers in the car are liable to receive an unwelcome shower bath. The best practice is to locate vertical cylinders in a separate well so located that the lifting ropes may be run directly to the elevator cars over sheaves at the top of the building.

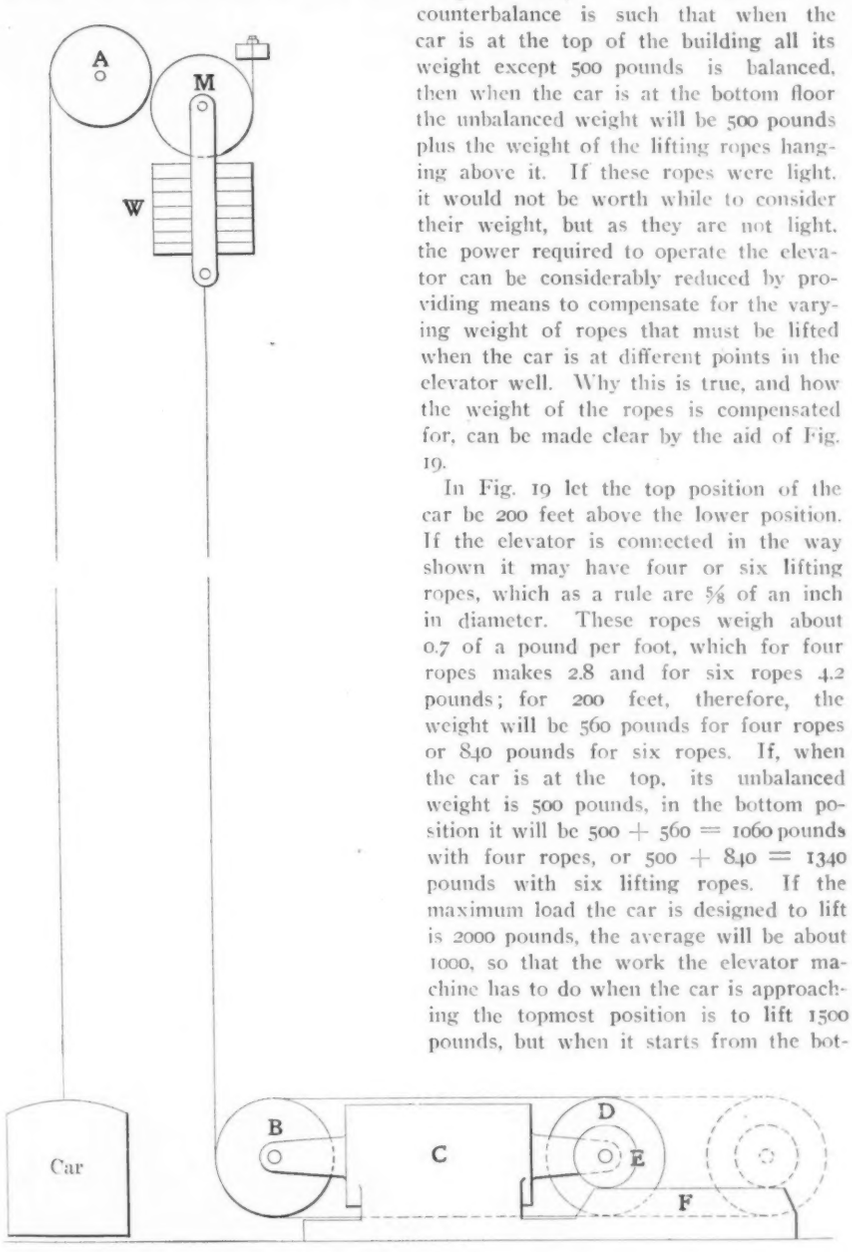


FIG. 18

COUNTERBALANCING THE LIFTING ROPES OF ELEVATOR CARS

When an elevator car is at the bottom of the well, the weight that has to be lifted comprises that of the car, its load and the ropes hanging in the elevator well above the car. When the car is at the top of the building, only the car and its load have to be counterbalanced. This being the case, it is evident that if the counterbalance is such that when the car is at the top of the building all its weight except 500 pounds is balanced, then when the car is at the bottom floor the unbalanced weight will be 500 pounds plus the weight of the lifting ropes hanging above it. If these ropes were light, it would not be worth while to consider their weight, but as they are not light, the power required to operate the elevator can be considerably reduced by providing means to compensate for the varying weight of ropes that must be lifted when the car is at different points in the elevator well. Why this is true, and how the weight of the ropes is compensated for, can be made clear by the aid of Fig. 19.

In Fig. 19 let the top position of the car be 200 feet above the lower position. If the elevator is connected in the way shown it may have four or six lifting ropes, which as a rule are 5/8 of an inch in diameter. These ropes weigh about 0.7 of a pound per foot, which for four ropes makes 2.8 and for six ropes 4.2 pounds; for 200 feet, therefore, the weight will be 560 pounds for four ropes or 840 pounds for six ropes. If, when the car is at the top, its unbalanced weight is 500 pounds, in the bottom position it will be 500 + 560 = 1060 pounds with four ropes, or 500 + 840 = 1340 pounds with six lifting ropes. If the maximum load the car is designed to lift is 2000 pounds, the average will be about 1000, so that the work the elevator machine has to do when the car is approaching the topmost position is to lift 1500 pounds, but when it starts from the bottom position the load that must be lifted is 2060 pounds if there are four lifting ropes, or 2340 pounds if there are six ropes. It will be evident that if the weight of the ropes could be eliminated, about 27 per cent. of the power could be saved if there were four ropes, or about 36 per cent. if there were six ropes. As a matter of fact the saving would be more, for, as may be seen from Fig. 19, when the car reaches the upper position, the traveling sheave *M* will be in the lower position *N*, and the length of the ropes from *G* to *H* will be added to the weight of the counterbalance; therefore, this weight would have to be deducted from the counterbalance in order not to have less than 500 pounds of unbalanced

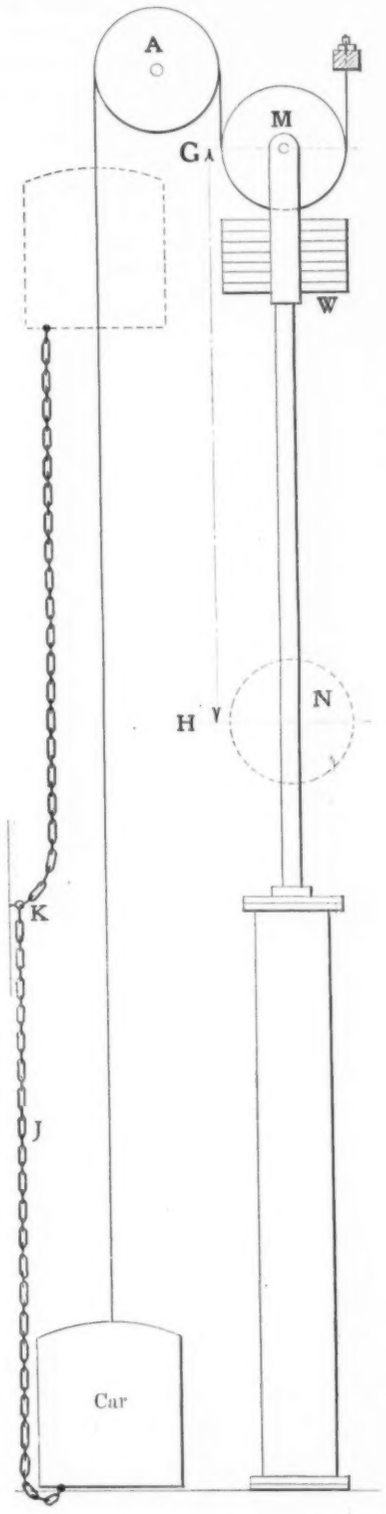


FIG. 19

weight when the car is in the top position.

The weight of the ropes cannot be eliminated, but the same result can be obtained by using a chain J secured to the side of the elevator well at a point K a trifle above the middle point, the other end being fastened to the underside of the car. When the car is at the bottom, the chain hangs from the point K, but as

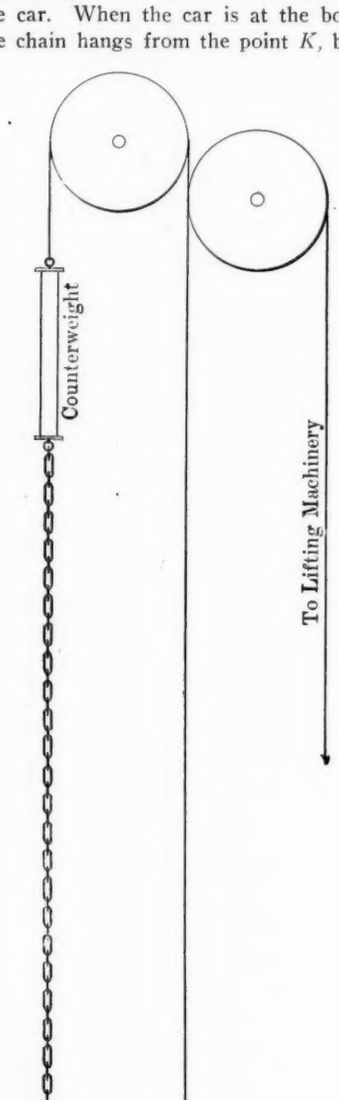


FIG. 20

the car ascends, part of the weight of the chain hangs on the car, and when the car reaches the top position all of the weight of the chain will hang from it. Now to make this chain compensate for the varying weight of the lifting ropes all that is necessary is to make it weigh as much as the 200 feet of ropes that hang in the elevator well when the car is at the bottom plus the length of ropes from G to H when the car is at the top. If the elevator is geared two-to-one, as in Fig. 19, the distance from G to H will be 100 feet, or one-half the rise of the car. For a three-to-one gear, G—H will be equal to one-third the rise of car, and so on for any other gear.

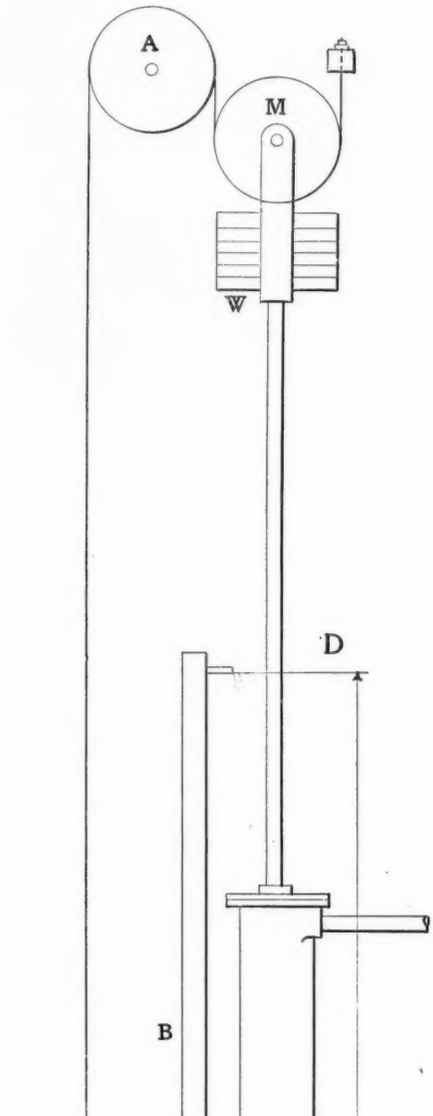


FIG. 21

When the compensating chain is secured to the bottom of the car, its weight acts in opposition to that of the counterbalance, and the latter must be increased in weight sufficiently to compensate for the weight of the chain. The chain can be attached to the underside of the counterbalance in some cases, and then its weight acts with the counterbalance so that the latter can be reduced in a corresponding amount.

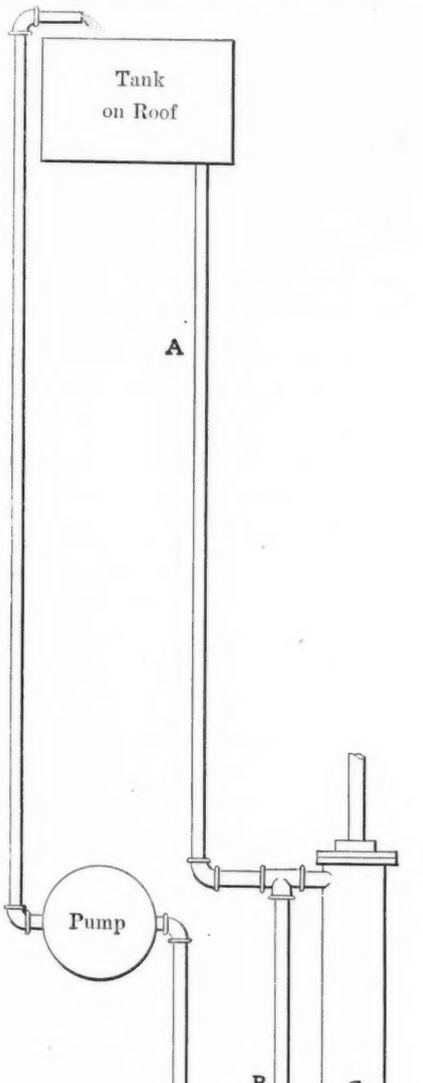


FIG. 22

The best way to connect the compensating chain is to attach one end to the underside of the car, and the other to the counterweight, as in Fig. 20. With

this arrangement the weight of chain can be reduced to one-half, because when the car is at the bottom of the well the chain hangs on the counterweight, and when the car is at the top, the chain hangs from the car. In practice, the chain is connected with the car in almost every case, because there is not sufficient space in the path of the counterbalance for it to hang freely without danger of being caught.

In a horizontal elevator with an independent counterbalance, as shown in Fig. 16, the counterbalance ropes add their whole weight to the counterbalance when the car is at the top of the building, and their whole weight to the car when the latter is at the bottom; hence, the weight of chain required to compensate for the counterbalance ropes is double the weight of these ropes, and to compensate for the lifting ropes a weight equal to that of those ropes must be added.

In practice, usually two compensating chains are used, and although they function perfectly and never give trouble, they are somewhat unsightly, and on that account a method of compensating for the varying weight of the ropes by means of a column of water has been used in some cases. This arrangement is illustrated in Fig. 21. When the piston descends in the cylinder *C* the water under the piston is forced up into the stand-pipe *B*, and this exerts a back pressure that increases as the piston travels downward. When the piston is at the top of the cylinder, the car is at the bottom of the building and, therefore, the weight of the ropes has to be lifted. At this time, however, the water in the stand-pipe *B* is at the level *E*, so that there is no back pressure, and as a result the lifting force is equal to the full pressure of the water admitted above the piston. When the car is at the top of the building the piston is at the bottom of the cylinder, and the water in the stand-pipe is forced up to the level *D*, so that there is a back pressure acting on the underside of the piston equal to the head *E*—*D*.

To make this arrangement compensate fully for the varying weight of the ropes all that is necessary is to make the internal diameter of the stand-pipe *B* such that when the piston reaches the lowest position in the cylinder the weight of the water in it will be just sufficient to offset the weight of the ropes. It will be noticed that this back pressure pushes upward against the piston and thus holds up a certain portion of the counterweight, so that a weight can be used that would be sufficient to overbalance the car entirely when the latter is at the top of the building if it were not for the assistance given by the back pressure of the water in the stand-pipe. It might be supposed that to obtain the exact weight of water required it would be necessary to make the stand-pipe of cast-iron sections and bore them accurately to the proper size, but in practice it has been found wherever this arrangement has been installed that ordinary steam pipes can be used that are near enough to the proper diameter to compensate for the weight of the ropes within less than four or five per cent. The water supply under the cylinder is kept up to the proper amount by what leaks past the piston, and in fact an overflow is provided at the upper level in the stand-pipe to carry off the excess. This arrangement, of course, cannot be used with a circulating pipe, and it may also be mentioned, that the lower level *E* cannot be more than about 32 feet below the underside of the piston when the latter is in the highest position.

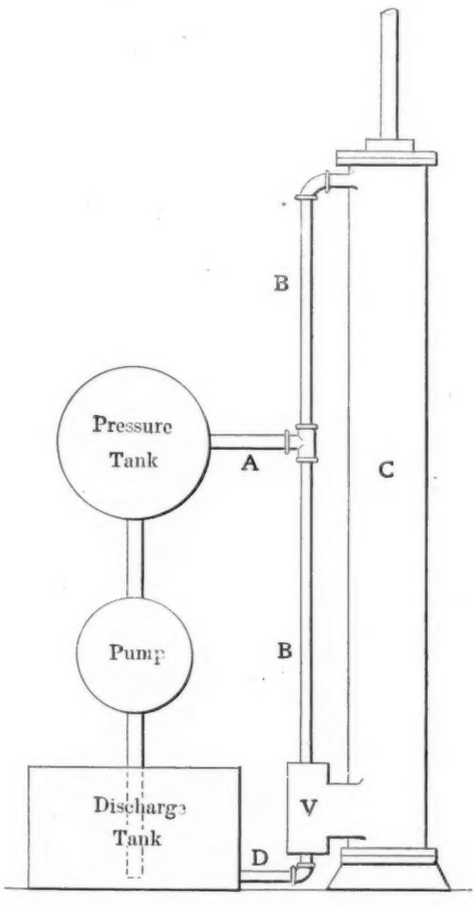


FIG. 23

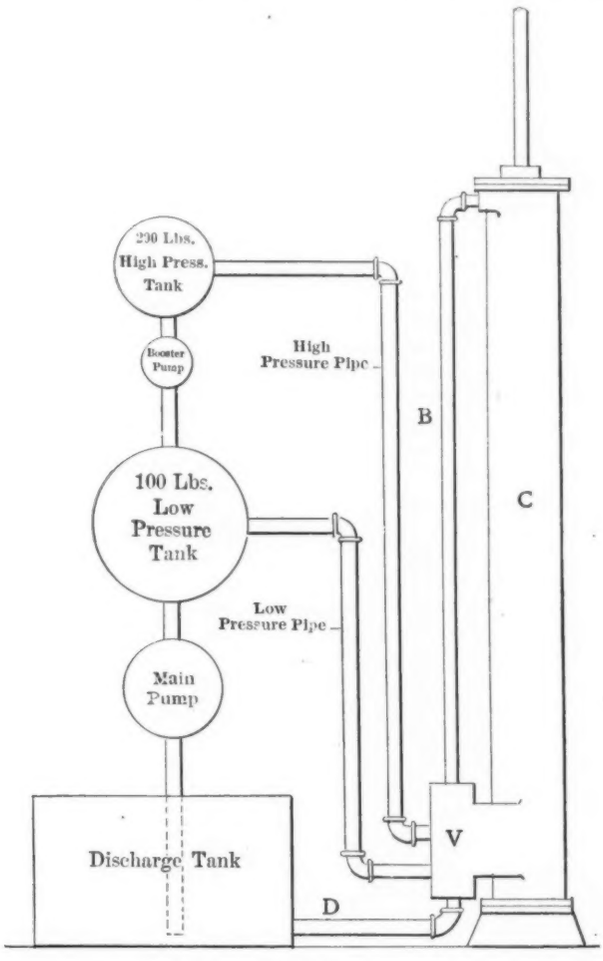


FIG. 24

GENERAL ARRANGEMENT OF HYDRAULIC ELEVATOR SYSTEMS

There are several ways in which hydraulic elevator systems are arranged, and these are shown in the sketches from Fig. 22 to Fig. 26. The first one

is what is called the gravity system, and is the type first used. A tank is placed on the roof of the building and the water is pumped into this to obtain the necessary head to actuate the piston in the lifting cylinder *C*. Water passes from this tank through the pipe *A* to the upper end of the cylinder, and forces the piston downward and the car upward. On the return stroke of the piston the water circulates through pipe *B* from the upper to the lower end of the cylinder. On the next down stroke, the upper end of the cylinder is again filled from the pipe *A*, and the water under the piston passes out through the pipe *D* to the discharge tank, whence it is raised to the roof tank by the pump. The pump is stopped and started automatically by a float in the roof tank, which closes the starting valve of the pump when the water reaches the high level, and opens it when the water is drained to the low level.

The system shown in Fig. 23 is known as the pressure-tank system, and differs from Fig. 22 only in having a pressure tank in place of the roof tank. With the gravity system the pressure is generally between 25 and 40 pounds per square inch, according to the hight of the building. With the pressure-tank system the range is between 50 and 200 pounds. In the pressure-tank system the pump is controlled by a pressure regulator that is connected with the pressure tank.

Fig. 24 is a system that has been installed in a number of large buildings by the Otis Elevator Company, and is known as the double-power system. There are two pressure tanks, one carrying about double the pressure of the other. The main pump delivers water into the low-pressure tank, and a booster pump draws from this tank and delivers into the high-pressure one. The elevator cylinder operating valve is made so that when opened part of the way it draws water from the low-pressure pipe only, and when opened further it draws from both the low-pressure and high-pressure pipes. If the load in the car is light, the desired speed can be obtained with the low-pressure water alone, but if the load is so heavy that the low pressure will not give the desired speed, the valve is opened further, and then high-pressure water runs in to help the low pressure and gives the proper velocity of car. A check-valve is included in the low-pressure pipe to prevent the high-pressure water from running back into the low-pressure tank; but unless the load is very nearly up to the maximum, water will be drawn from both tanks. This system was devised in order to make the power used by the elevator more nearly proportional to the loads lifted. When the low-pressure water is used, the power consumption is about one-half that when the high pressure is used. For loads greater than one-half the maximum, the power required is more than one-half but less than the whole amount, unless the load is very near to the maximum, in which case all the water used will be drawn from the high-pressure tank.

The three systems illustrated in Figs. 22 to 24 are of the low-pressure type, and can be used to operate one or any number of elevators.

THE GAS POWER SITUATION

Continuation of the Discussion Begun at the Recent A. S. M. E. Convention

At the December convention of the American Society of Mechanical Engineers, the discussion of Mr. Junge's paper on "The Evolution of Gas Power" was curtailed by reason of time limitation, both Dr. Chas. E. Lucke, of Columbia University, and R. E. Mathot, of Belgium, having been cut off by the time limit before they had fairly warmed up to the subject. We present below the remarks of these gentlemen as completed after the adjournment of the convention.

BY DR. LUCKE

Anyone, on reading this paper, would be apt to get a high opinion of gas engines for power generation, and, I think, too high an opinion. Gas power, to be worthy of consideration by power-plant engineers, must be considered in large installations by engines of large size, and should not be discussed for small sizes at all. Large gas engines have peculiarities and troubles not possessed by small engines, and comparison of steam engines and gas engines becomes rather more difficult in the larger sizes than in the smaller ones. I wish, therefore, to examine this question of gas vs. steam power, raised by Mr. Junge, and in the light of my presentation to point out a criticsm of this paper, and perhaps venture on a prediction.

I will divide the subject into headings for the examination of the problem:

First: "The theoretical possiblities of a perfect gas used in various cycles vs. steam used in its best cycles as a method of transforming heat into work." Such examinations on mathematical and thermodynamic grounds have been made many times, and they have always proved the superiority of the perfect gas cycle over any steam cycles that can be devised. Therefore, on this point I think I may say, without fear of contradiction, that the perfect gas cycle is better, and a more efficient means for transforming heat into work, than any vapor cycle in which the latent heat necessarily rejected is so high or in which the difference between total heats at high and low pressures is so small. This would seem to give the gas engine a superior position, and it is along these lines that most of the discussions in print on the superiority of the gas engines are based.

Second: "The mechanism for carrying out the cycle in a practical machine." On this point I can easily imagine an endless discussion. There are, however, one or two considerations that seem to me more prominent than others, and more important at this time, because not generally recognized. The gas engine, in its modern form, that is to say the form in which it appears in the large sizes, has been through a process of development of only about ten years. We have today large gas engines that will run. Ten years ago we did not. We have not today, however, a specially designed gas engine for each particular set of circumstances under which gas engines have to work. Builders of gas engines have, therefore, taken this single gas engine that would run under certain conditions, not always clearly defined, and have sold it to perform any kind of work under any other conditions, equally indefinite, and the engine has frequently failed as a result. We are today just beginning to recognize the importance of adapting the gas-engine mechanism to circumstances and conditions, and are still discovering what conditions affect its operation and what do not. When all of these conditions affecting the operation have been discovered and engineers shall have been educated to use this knowledge in designing proper mechanism, then and then only will we have special gas engines that can fairly compete with steam engines. The steam-engine advocates are apt to criticize the gas-engine advocates, and the gas-engine advocates are apt to be too sure of the results of the gas engine. This situation is directly a result of either ignorance of the importance of operating conditions and peculiarities of design, or a deliberate ignoring of this knowledge, which can only be attained by cost experiments too costly by far to be ignored.

Third: "The availability of the fuel." In the early days only gas fuel was burned; later on, vapor of the oils; still later, by-products, such as coke-oven gas, and lastly, but most important, gas made from coal in producers. It may be fairly said, therefore, that in the question of the availability of fuel, the steam engine has no position of superiority over the gas engine with the bare possibility of the caking bituminous coal in producers as the one exception.

Fourth: "Adaptability of the gas engine to the work it must do."

Fifth: "The skill or cost of the operating labor."

Sixth: "The first cost of the machine or plant."

Seventh: "Cost of maintenance and repairs."

Several other items of a similar nature can be added to this list of points of view from which the comparisons may be made, but all of them hinge upon the one question of "the design of the mechanism of the gas engine" to enable it to do a special service under all conditions imposed. If it should appear that the mechanism can be made as reliable, as cheap, as easily maintained, as adaptable to the work, etc., in the gas engine as in the steam engine, the gas engine would undoubtedly have a superior place. Unfortunately, this has not yet been proved, and the importance of it is even not recognized by some of the gas-engine builders. The steam engine has been through such a process of development for many, many years, and it is not yet finished.

Ever since the time of James Watt, we mechanical engineers have been designing steam engines and are still designing them, every day a different one. In other words, we have found it necessary to especially adapt each particular steam engine to the kind of service it has to perform, and to the conditions under which it must work. How different the engines of the locomotive from those of the steamship, and how different these from the engines of a large central power station. How different are small steam pumps from the large steam pumps, and a hoisting from a pumping engine. How different the high-speed steam engine from the slow-speed steam engine; the steam engine using low pressures from that using high-pressure steam.

We have today no gas engine specially adapted to pumping water, no gas engine fitted for driving ships, no gas engine generally recognized as the one for close regulation, no gas engine specially adapted for mill work, as distinguished from electric generation, no gas engine built especially for long life, no gas engine for power purposes especially distinguished for its small space per horsepower, nor one adapted to producer gas as distinguished from blast-furnace gas, or to dirty gas as distinguished from clean gas. In short, we have not only not yet designed special gas engines for special conditions but are only now beginning to realize the necessity for so doing. The failure to recognize the necessity for so doing is the cause of much loss of money and much loss of prestige of the gas engine in the power-plant world. I know of only one company building large gas engines, out of a possible list of a dozen or more, that has made any money; practically all of the others have lost money in the business. I know of a great many gas-power plants and gas engines that have been rejected for failure to fulfil contract requirements, and which have come into the courts for public airing.

This loss of money and these failures, together with loss of prestige, and by the loss of prestige, business, which is its consequence, are due solely to one thing, and that is ignorance of the limitations of the gas-engine mechanism. The builder of the gas engine did not know how to make it particularly adapted to the work. His knowledge was, in many instances, derived from a few experiments in his shops, or, perchance, from drawings and information obtained from Europe, the home of the gas engine. At this stage he was probably approached by a purchaser, who had read in the papers of the wonderful performance of the gas engine, the machine that could produce a horse-power-hour on a pound of coal of any kind—any time and all the time. It was with such an idea as this that the prospective purchaser approached the sales department.

The builder, having spent so much money on experimenting, trying to get his machine to run and having finally succeeded in making it run, was faced with the demands of the purchaser for a guarantee of 10,000 B.t.u. per horsepower-hour. He may not have ever been able to get as low as 15,000; he may not have ever tested his engine at all because of the cost of large gas meters. He may have been dependent upon the same published reports himself, and in his anxiety to get back his money he gave into the demands of the purchaser. The engine failed, doing much harm to his business, beside the immediate loss of money.

Now, the point I am making is not that gas engines are going to fail and continue to fail, but that these contracts were made on insufficient information on the part of the builder and unfair demands on the part of the purchaser, who, knowing nothing of the subject, allowed himself to be controlled by the public press. The purchaser did not know what was fair to demand, except in accordance with what he had read, much of which was false. The builder, either through lack of time, lack of sufficient capital to experiment properly, indifference, lack of able designers, or refusal to take the advice of good engineers, did not know what his engine could do, or did not care.

When the public shall have been educated to know what it is fair to demand of the gas engine, and to recognize what a gas engine can do, and when at the same time the builders of gas engines shall have recognized the importance of employing the best talent available to design their engines to meet special conditions, and shall take the advice of these experts, as to the importance of recognizing limiting conditions, then will the gas engine take its place properly beside the steam engine, and not before.

Mr. Junge in his paper scarcely mentions this point of view, which seems to be so vital to the question. He mentions some high thermal efficiencies, I believe impossibly high ones, but these are of absolutely no use in an engine when at times and under certain circumstances, that engine will not run. To the public purchasing gas engines or any other sort of engines for power purposes, I appeal: First, to recognize that the gas engine is at present a factor to be considered in every power proposition, and that it is not to be ignored in favor of any steam turbine, water power or other system, because, perchance, it is not so familiar; second, to recognize that the gas engine cannot do everything, especially when it is in the one-design form, and that what it can do should be best known to its builders and not to the writers of some magazine article; third, to keep the gas-engine builders informed of your special requirements, and invite bids on every power proposition, whether it seems likely they can meet it or not, and in issuing this invitation meet the builder half way by not imposing utterly ridiculous conditions.

To the builders of gas engines, I make an appeal as earnest as the one I make to the purchasers of this class of machine: First, employ the best men on general power-plant practice that your money can secure, and consider that man most valuable, who with the above information also knows the peculiarities of your engine and that of your competitor, with the limitations of both; second, seek to fill the special needs of purchasers without forcing on the public an engine that any good and competent engineer can plainly see is not adapted to the work; third, properly experiment for the purpose of determining what modifications of design and detail must be made to meet special service conditions and, when once determined, execute them; fourth, co-operate with purchasers of gas engines or power plants of any sort by exchanging freely all information on requirements and performance, and give up at once the hermit-like attitude of isolation and secrecy heretofore so common.

BY R. E. MATHOT

Following is an abstract of R. E. Mathot's discussion of Mr. Junge's paper:

The iron and steel industry is the one which has mainly caused the rapid growth of large gas engines and Germany has kept at the head of the development owing to the importance of its metallurgical industries. In that country it has been recently stated that among fifty smelting works actually at work, forty-two are already using, or have ordered, large engines for dealing with the gas generated in the blast furnaces or smelting ovens,

or coke ovens. This represents 350 units that give an aggregate output of about 400,000 horse-power, the largest of these plants being 35,000 horse-power, while there are fifteen works using plants of 10,000 to 12,000 horse-power. In some of them only provision is made to work with producers in case of need, to keep the plants at work.

In collieries and coke-oven works, the competition by internal-combustion engines against steam engines is difficult on account of the great number of old ovens from which the available heat can only be used in firing steam boilers. In these installations, however, the number of engines at work or in contemplation amounts to twenty or twenty-five, aggregating a total output of 35,000 to 40,000 horse-power. Almost all of the engines used in both smelting works and collieries are of the double-acting form, some of the two-stroke-cycle and some of the four-stroke-cycle type, the latter being, of course, the more largely applied on account of their higher efficiency.

An ordinary blast furnace of a daily output (24 hours) of 100 tons of pig-iron liberates about 315,000 cubic feet of gas which is available for motive power and is of an average heat value of 110 B.t.u. per cubic foot. This quantity of gas generates, in steam plants, about 2500 horse-power while it gives with gas engines 4200 horse-power, or about 70 per cent. more power. Such figures, of course, may not be expected unless the plants are provided with modern improved features, among which the most important is means for cleaning the gas, which has recently received careful attention from manufacturers of large gas engines as well as from the users themselves.

To get rid of the general impurities that the gas contains, such as dust, tar and chemicals, that would be detrimental to the good working of the engines, as well as in view of reducing the temperature of the gas before delivery to the cylinders, thorough cleaning, scrubbing and cooling are necessary. These operations are effected by means of fans, rotary washers, or the like, that involve a water consumption ranging from 0.25 to 0.40 gallon per 100,000 cubic feet of gas. The content of dust can by this process be reduced to from 0.3 to 0.2 of a grain.

The power required for operating the fans and the washers, depending on the systems as well as the amount of impurities to deal with, ranges from one and five-tenths to four horse-power per 100,000 cubic feet of gas; that is, about three per cent. of the power generated from the gas. With respect to the engines themselves, the cooling water required per hour per brake horse-power for pistons and piston-pins is from 2 to 3 gallons, and for cylinder jacket, etc., from 7 to 10 gallons. Lubrication in a good engine can be effected with one to one and a half grams of oil per brake horse-power-hour.

In view of meeting as closely as possible one of the unquestionable advantages of the steam engines with which we have to compete, we aim to build our gas engines with such dimensions that they afford a large margin of power, and although our best engines are capable of mean effective pressures of 90 and 95 pounds per square inch, we rate what may be called the constant working power on the basis of about 75 pounds. Our good four-stroke-cycle engines show an average thermal efficiency of 30 per cent., relating to the effective horse-power. This corresponds to about one brake horse-power-hour on 8500 B.t.u., which is realized in small single-acting engines, as well as large double-acting engines, when working at normal load.

In this connection let me observe that I have heard of certain American inventors who claim to reach much higher efficiencies with their plants. Tests made recently on small producers purport to show for the complete plants, including generator and engine, 36 to 40 per cent. thermal efficiency, after a trial of—two hours! Others state that their producer has 100 per cent. efficiency, and still others would convert nitrogen into available power. I know that from both the industrial and commercial points of view, American conditions are rather different from those in Europe, but I can hardly believe that the differences extend to the nature of "your" nitrogen, and ours on the old continent has remained obstinately inconvertible. Surely the question of gas power is interesting enough of itself, without having to advocate it in sweeping assertions that are liable to be more detrimental to the good fame of this motive power than modest and unquestionable truth.

Now, allow me to select some results of tests made by some of our leading authorities and some taken from 200 trials that I have been called to make myself.

(1) Trial of ten hours with prony brake on a 40-brake-horse-power suction producer and single-acting engine of the Maschinenfabrick Winterthur:

Consumption per brake horse-power at full load, 0.7 pound; consumption per brake horse-power at half load, 0.94 pound per hour of anthracite coal of 13,850 B.t.u., including ashes and moisture.

(2) On a similar engine I had already found with illuminating gas a consumption per brake horse-power-hour at four-fifths load of 17.6 cubic feet of gas, referred to 0 degrees Centigrade, and atmospheric pressure, of a heat value of 545 B.t.u. (lower value).

(3) Test made by Professor Schröter on a Güldner engine and producer; piston bore 250.6 mm., stroke 400.3 mm.

It should be remembered that the foregoing figures show low mechanical efficiencies because they relate to engines provided with very heavy fly-wheels in order to obtain extreme regularity of rotation.

Accurate figures on the consumption of large double-acting engines are unfortunately rather seldom obtainable, those engines dealing with such large quantities of gas that gas holders of sufficient capacity are rarely available for a reliable test. I may mention, however, a trial

TESTS WITH DIFFERENT FUELS ON NÜRNBERG SINGLE-ACTING ENGINES.

Plant.	No. 1.	No. 2.	No. 3.
Fuel.	Anthracite.	Coke.	Illum. Gas.
Working load horse-power	107.4	110	152.8
Consumption per indicated horse-power-hour in engine	0.78 lb.	0.93 lb.	15.7 cu.ft.
Heat consumption per indicated horse-power-hour in suction producer; B.t.u	10,850	10,840	
Thermal efficiency of producer	80%	75%	
Heat consumption in the engine per brake-horse-power-hour; B.t.u	6750	6300	6200
Mechanical efficiency of engine	80%	80%	78%
Thermal efficiency of plant, relating to indicated horse-power	36.3%	38.3%	36.6%

OWNERS AND LOCATIONS OF THE PLANTS:

No. 1 Royal foundry of Wurthemburg (Wasserfllingen).
No. 2 Imperial Post Office at Hamburg.
No. 3 Municipal Electric Station of Greifoland.

TEST WITH ILLUMINATING GAS.

Load Ratio.	Rev. per Minute.	M.E.P.; Kg. per Sq. Cm.	Load; I.H.P.	Heat Value; Cal. per Cu. Meter	Consumption per Hour per I.H.P. Ref. to 0° C.; 760 mm. Bar.	Thermal Ind. Efficiency, Per Cent.
½	213.9	4.48	21	4420	0.3975	33.9
¾	212.8	6.71	31.3	4410	0.347	38.8
full	214.5	8.06	37.7	4430	0.3435	39
full	210.7	7.76	35.9	4440	0.3145	42.7

TEST WITH SUCTION FUEL GAS FROM ANTHRACITE COAL.

Load.	Rev.	M. E. P.		I. H. P.	Heat Value.		Gross Fuel Con. B. h. p.-hour.		Thermal Efficiency Indicated.
		Kg.	Lbs.		Cal per Kg.	B.t.u. per Cu. Ft.	Kg.	Lbs.	
Full	210	7.6	108	34.9	7780	13.878	0.336	0.739	28.5

witnessed by the engineers of both the makers and the users on a double-acting four-stroke-cycle engine of 600 horse-power supplied by Ehrhardt & Sehmer, one year ago, to the Königlische Berginspection at Heinitz Saarbrück, Germany. After four months of constant work and without previous cleaning, this engine was tested with coke-oven gas ranging from 350 to 370 B.t.u., and showed an economy of 8100 B.t.u. per brake horse-power-hour. The mechanical efficiency recorded, with the power under consideration, was 83 per cent. The engine was a new one and was tested under normal load at 150 revolutions per minute. The principal dimensions are: cylinder bore, 620 mm.; piston stroke, 750 mm.; diameter of rods, 170 mm. The load reached 520 kilowatts at the terminals of a three-phase dynamo mounted on the crank-shaft of the engine. It will be seen that the above figures show a th rmal efficiency of about 31 per cent. on the basis of brake horse-power and 37.5 for the indicated horse-power.

High efficiencies, smooth running and reliable working are all obtained by reason of the following features of design now applied by almost all European makers. The compression has been raised to 160 to 190 pounds in order to obtain reliable ignition of the very lean mixtures used for purposes of economy. High compression involves high temperature and we have therefore to design the combustion chamber to allow uniform cooling and free expansion of the cylinder head. We aim also to design the combustion chamber of such a shape that it affords the maximum volume with the minimum cooling surface and facilitates high velocity of flame propagation in the explosive mixture as well as thorough combustion without the sharp explosions which are of such detrimental effect in the old type of hit-and-miss engine now completely abandoned by our representative makers. In fact, whatever the quality of, or richness of the gas used, in spite of high compression, we aim not to reach initial explosive pressures above 330 to 360 pounds. This causes our engines to run smoothly, without pounding.

Governing is always effected by varying the mixture admitted at each cycle, whether by varying the quantity at constant ratio, or by varying the ratio of gas in a constant quantity of mixture or by combining both processes. The first method causes, of course, variable compression and, as a consequence, some loss of power due to partial vacuum in the cylinder at low loads, but in spite of this defect it has the advantage of giving the highest efficiency at every load because it results always in good combustion of the mixture, exploding in due time.

The second method, although being apparently less economical, holds certain mechanical advantages.

The third method, involving a combination of both systems of variable quantity and variable quality, is claimed by its few advocates to possess the leading advantages of the two former methods, without having their weak points. But the combined system leads to the use of somewhat complicated mechanical arrangements and its reliable operation might therefore be questionable.

The most rational course seems to consist in the selection of that one of the first two methods which suits better the character of the work the engine has to deal with.

In the case of high-speed engines supposed to run at a nearly constant number of revolutions, as for driving electric alternators, spinning mills and the like, the inertia of the principal reciprocating parts becomes an important factor of smooth working. The reciprocating masses should therefore be kept at a constant speed and the system of governing by variable quality should consequently be preferred, because it gives constant compression.

In the case of slow-speed engines such as are used for driving blowing plants, pumps, rolling mills, etc., which allow variations in the number of revolutions to the extent sometimes of 50 per cent., the system of governing by variable quantity with constant quality of mixture will answer the purpose, despite the variation of the compression.

All large continental engines are made of the double-acting horizontal type, and the extent sometimes of 50 per cent., the off steam engine with valves located at both ends of the cylinder has been adopted almost universally. The inlet valves at the top and exhaust valves at the bottom meet both constructional and working requirements in every respect. In this respect the engines of the Allis-Chalmers and the Westinghouse Companies in this country are quite up to date. The question whether their side crank is better than our center crank will be solved by future experience though, nowadays, it meets better the American requirements as to simplicity and facility of erection, which are due to lack of training of your young engineers.

I rather agree that our European engines are too complicated and am quite confident that Americans will before long design large engines that will be much more simple and reliable, thanks to their remarkable faculty of adaptation and their ingenuity.

Modern Gas Producers

It is hardly necessary to point out that gas producers must be made to produce a regular quality of gas. In small producers of but little hight such a demand entails the use of a fuel that will not hang up and form caves and arches and blank corners. The rectangular producer is obviously incorrect, because its square corners are apt to become simple up-flues through which the air will travel direct to the top of the fuel bed, and will there burn any of the combustible gas which comes through the fuel. But any shallow producer, even of circular form, will burn into hollows unless the fuel be of such a quality as will readily consolidate itself. Hence the almost universal employment of hard anthracite coal, which does not clog, lies loosely and readily falls under the influence of gravity. Bituminous coal demands producers of greater hight, so that the weight of the high column will at least consolidate the lower portions and insure the production of CO and not of CO_2. In a lime-fluxed producer, no doubt, the fluxing of the clinker-forming material will help considerably to prevent hanging up of the fuel; but hight, to give gravity effects, is essential, and if producers be of square form, they must have rounded corners of large radius so that the above bad effects of square corners shall not take place.

In coal there are many things which are burned when the coal is used for steam raising, yet are more valuable when saved unburned. Some of the things, such as tar and ammonia, can be saved when gas producers are employed. So far as the process of gas production is concerned, no fumes of any sort are turned into the atmosphere. The most destructive of the products from coal is the sulphur, which ultimately becomes converted into sulphuric acid. It is claimed that if a proportion of lime be mixed with coal, the sulphur will be absorbed by the lime; but against such advantage as this may be is to be set the awkward fact that lime tends to produce a clinker, which would spread upon the bars and choke the air spaces. But if lime in the form of limestone or chalk be used as a flux in the producer, it allows the clinker to be run out liquid, and no doubt causes an absorption of much of the sulphur of the fuel.

Such sulphur as passes away in the gas must, of course, ultimately be present in the engine exhaust, but the proportion will be very much less than that evolved from the fuel consumed to raise steam for an equal steam power. It is, however, suggested that the sulphur gas in the engine exhaust would eventually become sulphuric acid, and that such gas would best be purified and ultimately caused to sulphate the ammonia derived as a by-product of the producer. Could this be done the final exhaust gases would consist of CO_2 and nitrogen only.

However, these are matters for the manufacturing chemist to determine; but it certainly appears feasible commercially to put forth into the atmosphere nothing deleterious, save, of course, the carbon dioxide.—W. H. B., in *The Electrical Review*, London.

CORRESPONDENCE AND DISCUSSION

Correspondence upon Topics of Interest Relating to Power Is Especially Solicited, and Accepted Contributions Are Paid For—Answer Early in Month if Matter Refers to Preceding Issue

Raises Issue with Mr. Porter

As Chief Engineer and Superintendent of United States Public Buildings, and working under the Departments of the Treasury and Commerce and Labor, at the Port of New York, since 1899, the following, relevant to one of the building plants under my charge, seems called for, regarding an article headed "Engineering Reminiscences," by Charles T. Porter, in the December, 1906, issue of POWER.

In his article Mr. Porter gives neither names nor dates, and hence is very likely, innocently perhaps, to do grave injustice to the present, rather than to a past, generation. The following facts seem worthy of note:

(1) The present steam and electric plant in the United States Court House and Post Office, New York, was installed under my direction when I first came on temporary duty at New York in 1897. The venerable Charles T. Porter has never had anything whatsoever to do with said plant.

(2) For about six years or more prior to that time the United States took steam from the New York Steam Company, and current from one or the other of the local lighting companies, for use in said building.

(3) The electric power plant, which was destroyed by fire in, I believe, 1891, was not operated by the United States, but by persons employed by contractors for lighting, and I infer that it was with these employees that Mr. Porter had his experience. However, a statement from him as to the name of the engineer or postmaster with whom he had correspondence or relations would settle this question.

ALFRED BROOKS FRY.
Chief Engineer & Superintendent,
U. S. Public Buildings, Port of N. Y.

Automatic Engine-stops versus Fly-wheel Insurance

In the June number of *Insurance Engineering*, William H. Boehm had an article citing some fourteen reasons why a fly-wheel may explode; also table showing safe speed of various types of wheels. This article proved of great value to the makers of automatic engine-stops. There are numerous devices for stopping an engine automatically, or, rather, to take the steam away, in case of overspeed. The most of them are built and installed by the engine builders—some good, some bad, and, as Mr. Boehm says in the October number of POWER, some are worse than nothing.

There has been a great deal of time and money spent in getting a stop that acts entirely independent of the governing mechanism, and a record of lives and property saved by its use is one to be proud of. It is only within the last few years that automatic engine-stops have been perfected from the standpoint of the electrical expert.

I am sure of my ground, and I make the statement that the engine has not been built yet that will not race, under conditions that arise during the life of the engine. The common expression among engineers is that an engine races. The moment the fly-wheel bursts, he calls it "runaway." The engine builder has been known to say that he never knew one of his engines to run away, but will admit that she can and will race under certain conditions.

We will leave our readers to determine where racing leaves off and runaway begins. Now, can Mr. Boehm, or any engine builder, raise any objection to equipping an engine with a device that, to say the least, helps to prevent racing until the term changes to runaway? With perhaps the exception of Henry M. French, I have visited more fly-wheel explosions than any other man, and I can assert that I never saw a fly-wheel wreck that would not have been prevented by a good automatic engine-stop, and I have never seen one that had a stop where the wreck could in any way be attributed to the stop. It was through the neglect of the engine tender. It is rare to find an engineer who allows his stop out of commission; it is rare to find an engineer who does not speak in praise of his stop, or one that he would like to have. I make a great big distinction between engineers and the engine tenders we find, oh, so frequently, even in well equipped engine-rooms. It seems incredible that a firm or corporation will employ a man, calling him chief engineer (with the emphasis on Chief), who does not know why it is dangerous to run an engine blocked.

I have no fault to find with fly-wheel insurance, only in so far as it encourages the employing of incompetent engineers. The same is true of automatic engine-stops. The law in some States requires a man in the engine-room at all times while the engine is running. To enforce this law would necessitate increasing the corps of inspectors beyond the limits set by politicians.

From long experience in the engine-room, and in calling on engineers (average of four a day), talking with them at their meetings, where they are free to discuss these questions, I prophesy that within the next few years a very large percentage of engines running a generator will be equipped with a good automatic engine-stop before they are put in commission. Both steam and electrical engineers will indorse the above statement. Then the pioneers who spent money and time inventing a device to protect innocent people from fly-wheel explosions will be praised. Sweet, Locke, Heminway, French and others, men and women in our large factories will rise up and call you blessed long after the men who would take away this protection in order to increase fly-wheel insurance business are forgotten.

F. S. PALMER.
Cleveland, O.

Mr. H. P. Moorrees Replies to Mr. Hellmer

I note R. Hellmer's answer in the December issue to my criticism in November POWER of Mr. Hellmer's translation of Professor Riedler's paper. I also noticed the editorial "Steam Turbines with Millions of Blades." In regard to the former I wish to call your attention to my original copy, wherein, I am sure, I did not mention a 100-kilowatt machine, but a 1000-kilowatt machine; also the figures as to steam consumption are not the ones given by me.

As to the "Millions of Buckets," I would say that it is easy to make a Parsons turbine with so many buckets—just build one that is big enough. But that was not the question at issue. You asked me: "Did Professor Riedler say so in his article?" and to this I say: No!

In conclusion I wish to say that Mr. Hellmer misinterpreted the meaning of my article entirely. It was meant in the defense of Professor Riedler, who was called an enthusiast, etc. Mr. Hellmer's article called out that editorial, and was therefore responsible for it. I intended to be understood as meaning that the spirit in which Professor Riedler gave his lecture was not such as warranted the criticism

made of him, whoever was the cause of it.

New York. H. P. MOORREES.

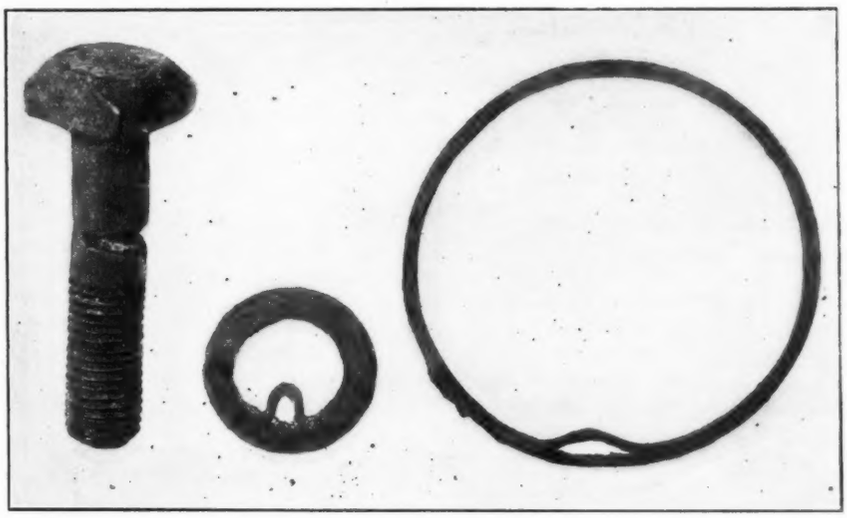
Effect of a Steam Leak—Defective Tube and Pipe

I send you herewith three exhibits.

First, a bolt worn away by a steam leak in a cast-iron flange under 70 pounds pressure. The leak was difficult to get at and went a long time before repairs were effected, and the job was worth doing, as the weakened bolt shows.

Second, a 4-inch tube which was taken from a water-tube boiler. We struck the defect with a Weinland tube cleaner and were perplexed and cut the tube out. I believe it would not have hurt to leave the tube in, but then we would not have known what the bump was.

The third is a sample of hydraulic pipe, which we were putting into a new line, and the defect in which we were lucky enough to find before installation.



SEVEN-EIGHTHS-INCH BOLT CUT NEARLY IN TWO BY STEAM LEAK FROM FLANGE. "PIPES" IN HEAVY HYDRAULIC PIPE AND BOILER TUBE

FRANK MARTIN.

College Point, N. Y.

"Explosive Gas in Pistons"

In reference to R. W.'s letter on page 699 of the November issue, and the reply by Gus A. Janicke on page 797 in December POWER, I venture to suggest another probable explanation. The air originally inclosed in the piston would contain some moisture, and when the piston became heated in the engine cylinder, the moisture in the air would react chemically with the iron of the piston, producing oxide of iron and hydrogen. This explanation is suported by two of R. W.'s statements. He says that it was a pale bluish flame, which is characteristic of hydrogen. He also says that this gas is found even if the piston has only been used a few days. The hydrogen would be formed the first time the piston was heated hot enough. I assume, although I have never seen it so stated, that it is hydrogen which comes from molds when the cast-iron is poured into them, the red-hot iron coming in contact with the damp sand, forming black oxide of iron and hydrogen.

I hope that someone who has accurate knowledge on these points will be good enough to set us straight.

Sherbrooke, Que. H. V. HAIGHT.

"Fly-wheel Wrecks—Safety Device Fallacies"

In the article published in October POWER, page 625, under the above caption, I cited the fly-wheel wreck at the Mansfield Railway, Light and Power Company's plant as an instance in which a special speed-limit safety device had failed to prevent an explosion of the wheel. I explained that the failure of the speed-limit device was due to its being struck by the main driving belt of the engine, the belt on breaking having been hurled into the device, after which the engine raced and burst its fly-wheel by centrifugal force.

In the December issue of POWER, page 782, Mr. Heminway, of the Consolidated Engine Stop Company, takes issue with my statement that the engine raced. He claims that the safety device was disconnected at the time of the accident, that the engine did not race and that the wheel was broken by an overload produced by a short-circuit.

The view of the wreck reproduced on page 379 of the June issue shows plainly that the *rim* of the wheel burst, that heavy roof timbers (they were 12x14-inch) were cut in two, and that everything in the path of the flying fragments of the rim was demolished. The roof was split open from one end of the building to the other, and the side walls were bulged out. Fragments of the rim were hurled great distances.

It goes without saying that these rim fragments must have been hurled with terrific linear velocity in order to accomplish such work of destruction, or be thrown such distances. Inasmuch as the linear velocity of these rim fragments just *after* disruption was the same as their circumferential velocity in the rim just *prior* to disruption, racing *must* have occurred.

The claim that this wheel was broken by an overload is absurd. An overload produces a twisting action, the effect of which is to so reduce the speed of the wheel that when failure occurs the pieces are thrown little or no distance, often merely dropping into the wheel-pit, and doing little damage except to the wheel itself. Few belt-wheels fail this way, because, however great the overload may be, the maximum strain that it is possible for it to exert on the wheel is measured either by the strength of the belt or its resistance to slipping on the wheel. I recall but two belt-wheel failures from overload. In both instances the pieces were thrown less than the width of the engine-room or fell into the wheel-pit. No damage was done in either instance, except to the wheel itself.

Then, too, the effect of the twisting action due to an overload is to break off the spokes of the wheel *near the hub*. The picture shows clearly that not a single spoke was broken in that way. It shows every spoke intact, and that the rim, burst by centrifugal force, must have been hurled from the spokes with the terrific velocity due only to racing.

The claim that a short-circuit occurred is probably correct. But the overload produced by the short-circuit broke the *belt*, not the wheel. The belt, after breaking, was hurled into the governor and safety device. The engine, then free of its load and receiving steam at full boiler pressure, raced and burst the wheel by centrifugal force, as originally stated by me.

The claim that the speed-limit device was disconnected at the time of the accident may be true. I found both the governor and speed-limit devices out of commission. Whether the speed-limit device was disconnected before or during the accident has little bearing in this case. The facts are that the owners of the engine equipped it with a special speed-limit safety device for the express purpose of preventing a possible explosion of the wheel, and the explosion occurred.

No safety device is more simple or positive than the common lever safety valve; the pop safety valve is equally so. Every boiler is equipped with one or the other; many are equipped with both; practically all are insured and inspected, and still boiler explosions are continually occurring.

It is the same with fly-wheels. The

governor of the engine corresponds to the safety valve of the boiler. The one limits *steam* pressure, the other limits *speed* pressure (centrifugal force). Equipping the engine with a special speed-limit device is equivalent to supplying it with two governors. This again is equivalent to placing two safety valves on the boiler. The extra safety device is in either case an additional safeguard and a praiseworthy precaution, but in spite of them accidents occur.

We would not insure a boiler without a safety valve. We would not insure a fly-wheel on an engine without a governor. We recommend but do not require an additional safety valve on the boiler. We advocate but do not require an additional governor, or, what is the same thing, a special speed-limit device on the engine. But in neither case does the extra safety device take the place of inspection and insurance. The reason is that all such devices consist of mechanism as likely to get out of order as any other mechanism about the plant. The fallacy consists in placing implicit reliance in them.

In pointing out the fallacy of depending upon safety devices we do not mean to belittle the extra safeguard they give. There are, in fact, certain conditions under which we require an additional safeguard, as Mr. Hemenway probably knows.

One of these is the condition where it is necessary to run with the governor blocked in order to prevent the engine from shutting down under a heavy load. This condition exists in many plants and the practice of blocking the governor is dangerous. In such instances we naturally insist upon some special device that will take the place of the regular device purposely put out of commission.

New York. WM. H. BOEHM.

A Restricted Job of Wiring

The following is a problem that I came in contact with on my last job; perhaps it will interest other readers. A three-story brick building with four suites of rooms each on the second and third floors, two stores in the rear of the first floor and a bank in the front, is to be wired for lighting. All public and bank lights, fed from four circuits in all, are to be supplied through one meter, and each separate store and suite of rooms through separate meters, making a total of eleven meters and fourteen circuits.

It is mixed concealed knob and tube work, with the exception of the main line feeds, which are No. 6 stranded rubber-covered wire brought to the main-line switch in the meter closet in one-inch iron conduit. All circuits must leave this closet from a cut-out cabinet. The size of the closet that the cabinet and meters must all be located in is 23x42 inches, with a 10-foot pitch.

The meters are Westinghouse, size 6x7 inches.

This work must be done to comply with the underwriters' rules. How is is done? FRANK W. HUDSON.

Medford, Ore.

Grinding Pump Valves

A short time ago our duplex pump failed to force water and an examination showed that the valve seats were badly scored. These seats were expanded into the pump and beaded over so they could not be removed, and to grind them in place was the problem we had to solve.

I took a 1½-inch coupling, as shown at *J*, Fig. 1, and filled it with babbitt and then drilled it out for a ⅝-inch bolt *B*. This bolt I threaded nearly the full length. Four small bolts were driven into holes drilled in the lower end of the coupling as shown at *M*. A bar of iron ⅜x1 inch was shaped as shown in Fig. 2, bent as in Fig. 3 and placed at *W*, Fig. 1. The nut, Fig. 4, was made from ⅞-inch iron and shaped to enter the holes *N N* drilled in the bar, Fig. 2. This is shown in place in Fig. 3 and at *C* in Fig. 1. The nut was drilled and tapped out for a ⅝-inch bolt *Y*, Fig. 3, the outer end fastened to form a handle as at *R*, Fig. 1. Small holes ⅜ of an inch in diameter were drilled in the side, forming a circle around the coupling. They were spaced ¾ inch apart and are shown in the collar *W*, Fig. 1. I took an old metal valve and ground it good and level on the face. Cavities were sawed and chipped out to allow the four pins at *M*, Fig. 1, to drop into them, and by making this connection the body *J* could not turn without turning the valve. The lower end of the bolt *B* was screwed into the hole at the center of the valve seat where the stud holds the valve in place under working conditions. A good coat of lard oil and ground emery was placed between the valve and the seat to be ground, and the lower end of the body *J* was set up snug enough against the valve to allow the coupling to turn and bring a force on the top of the valve. The strap *W* was turned until the taper end of the bolt *C* came in line with a hole in the body *J* at *W*, when the ⅝-inch bolt was turned in at the handle *R* and this made it fast to the coupling. The bolt *B* was screwed into the seat tightly so that it could not turn. The handle *N* was wound back and ahead, which moved the body *J* and with it the valve. To change the position of the valve, the bolt *C* was backed out and screwed into another hole to put on more energy. The nut *J* was backed off, the coupling raised and the seat examined. It will be seen that by backing out the bolt *C* and putting it into other holes in the body *J* a full revolution can be made. The job was done and well done too, for the old pump throws water good and fast. Fig. 1 shows the tool in place; the valve chamber being an extension of the water cylinder. To grind the suction-valve seats the head was taken off, but the discharge seats were ground through a hand-hole 12x7 inches located as shown in Fig. 1. It is well to mention that the strap *W* is free to move up or down or turn around the body *J* when the bolt *C* is not in one of the holes *W*.

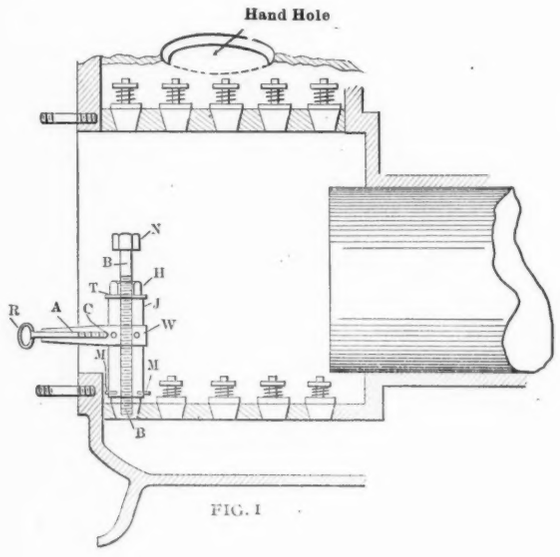


FIG. 1

FIG. 3

FIG. 4

FIG. 2

GRINDING PUMP VALVES

Cambridge, Mass. W. A. Dow.

Why the Difference in the Steam Line?

Inclosed find a pair of indicator cards from a 26 and 52x54 compound running at 78 revolutions per minute. The diagram from the high-pressure cylinder is taken with a 60-, and that from the low pressure with a 10-pound spring. The boiler pressure was 120 pounds, the pressure in the receiver 7 pounds and the vacuum 24 inches. I should like to have the ideas of some of your correspondents in regard to the difference in initial pressure as between the head and crank ends of the low-pressure cylinder.

Columbus, O. G. A. ACKERMAN.

Kindly enlighten the writer as to what caused the difference in the two low-pressure cards of the inclosed set of indicator diagrams. These two low-pressure cards were taken under apparently the same working conditions, same number of spring, same steam pressure, in the low-pressure receiver same revolutions per minute, cut-off alike, in fact, no change in the engine to all appearance and yet the one card carries the steam line much better than the other. The diagram on card No. 1 is that which the engine generally makes, but for about one-half hour the engine made that on card No. 2 without any alteration whatever and the engineer has since been unable to get a reproduction of card No. 2. The set was taken from a triple-expansion engine less than a year old in a modern lake carrier.

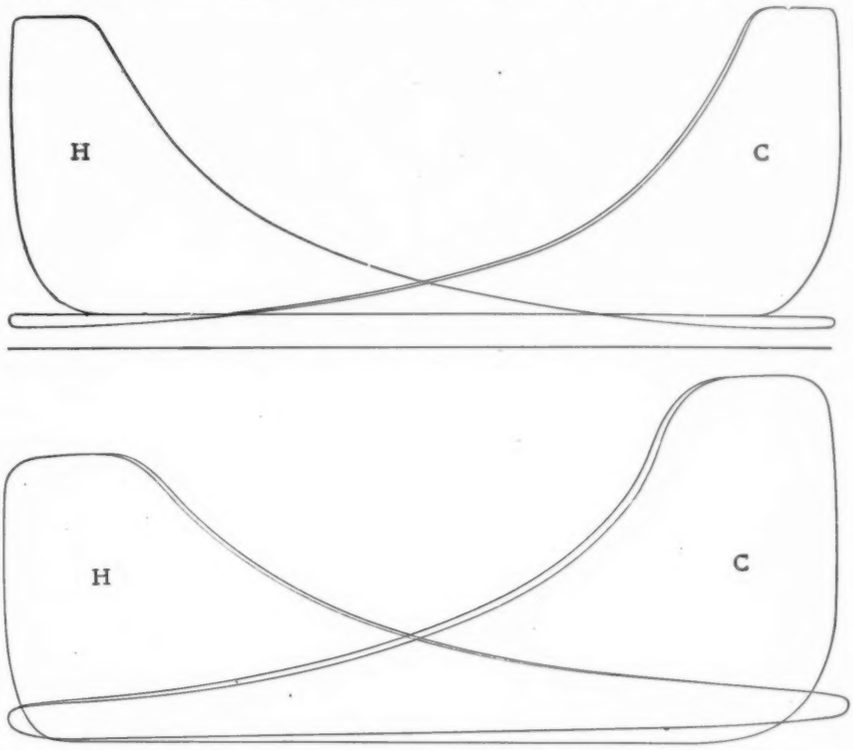
Conneaut, Ohio. GEORGE W. BACH.

Handy Horse-power Figures

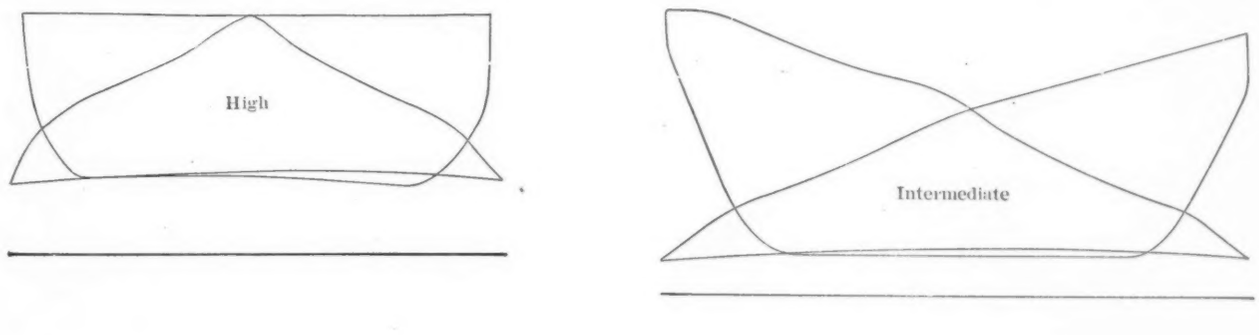
The number 33,000 is of course inseparably connected with horse-power in the minds of all who have to do with steam engines and other motors. No one forgets it. Now, if one will just keep in mind the number 303, it will often come in handy in the same connection—always remembering to put a dot over both the 0 and the 3, so as to make of those two digits a repetend.

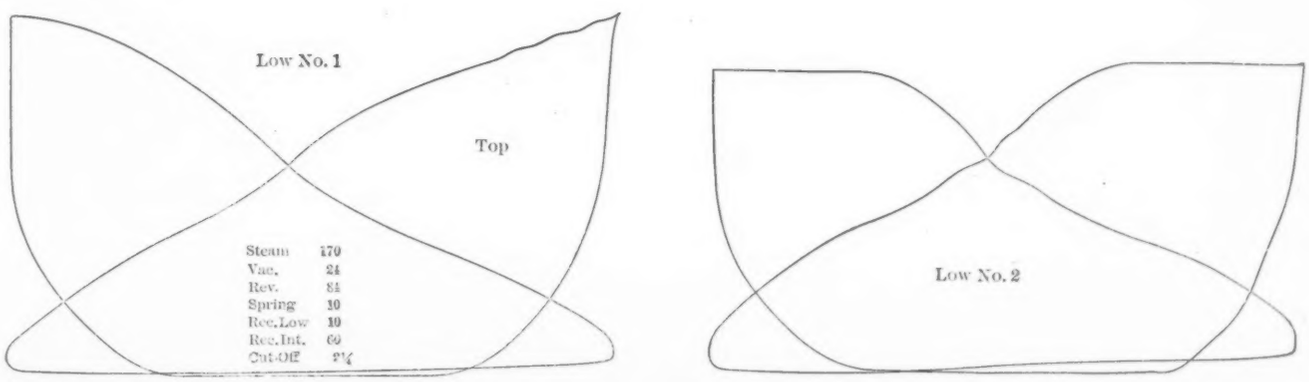
One pound per square inch mean effective pressure on the piston gives, at 1000 feet piston speed per minute, 3.03 gross horse-power for each square inch of net piston area.

One square inch of net piston area



MR. ACKERMAN'S DIAGRAMS





MR. BACH'S DIAGRAMS. WHY DOES THE STEAM LINE FALL IN NO. 1?

gives at 1000 feet per minute speed 0.303 gross horse-power.

One foot per minute piston speed gives, with one pound mean effective pressure for each square inch of net piston area, 0.0303 gross horse-power.

Applying these three simple rules to my formula

$$PAT \div 33{,}000 = H.P.$$

(which I prefer to Professor Marks' $PLAN \div 33{,}000$), we have for 10 pounds mean effective pressure 100 square inches net piston area (counting out half the area of the piston-rod, if there is no tail rod such as is found in so many European engines) and 1000 feet per minute piston speed = 33.303 gross horse-power, which is 3.03 times the mean effective pressure, 0.303 times the net piston area and 0.0303 times the piston speed.

ROBERT GRIMSHAW.

Hanover, Germany.

Water in the Cylinder

That all who have to do with the generation and use of steam are not acquainted with the cause of water in the cylinder is evidenced by the number of

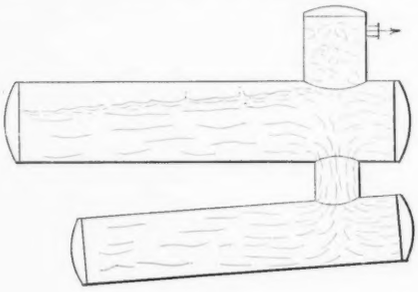


FIG. 1

accidents directly traceable to that cause. The water forms in the cylinder and, when a sufficient amount has accumulated, the pressure of the piston on such an inelastic mass naturally forces something to give way.

There are three reasons to explain the presence of water in the cylinder: First, the water may be carried over from the boiler; second, it may form in the steam-supply pipe; and, third, if a condenser is used, water may be drawn back from it through the exhaust pipe.

Sometimes the steam-supply opening is placed too near the surface of the water in the boiler, as indicated by Fig. 1. It is plainly evident that water will be more readily carried from a boiler designed after this fashion than from one like that shown in Fig. 2.

Then, too, the sudden demand on a boiler for more steam than its usual capacity will cause the formation of so-called wet steam. This danger of course increases with the excess requirement. For example, if we suddenly increase the demand of the engine for steam with a boiler giving, say, 90 pounds with a temperature of 160 degrees, then the pressure may drop to say 75 pounds, requiring a temperature of only 153 degrees. Since the water cannot so suddenly change its temperature, there is present an excess of 7 degrees, which will cause too violent

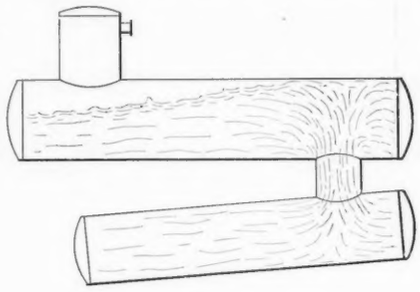


FIG 2

ebullition of the water, and the too rapid generation of steam will cause the carrying over into the steam pipe of small particles of water. This of course can be taken care of by a separator of sufficient capacity.

The presence of water in the steam pipe is often due to too great a length of

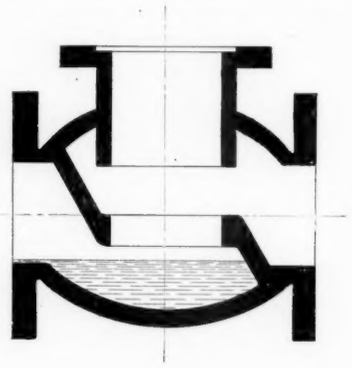


FIG. 3

supply pipe, or, to a lesser extent, a pipe of too small diameter. Long supply pipes should be avoided as far as possible, but when circumstances are such as to compel their use, they should be well insulated. Care should also be taken to avoid all pockets and to use correctly designed valves.

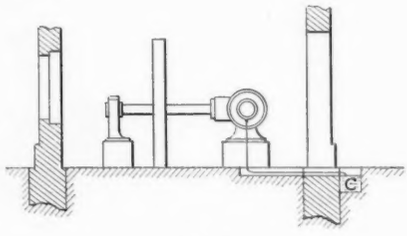


FIG. 4

In using a valve such as shown in Fig. 3, it is preferable to set it vertically rather than horizontally, or, if it is placed horizontally, the use of a pet-cock to drain the lower chamber is recommended. The supply pipe should also be inclined from the boiler to the separator and this latter should be placed as near the engine as possible.

With unjacketed cylinders water is formed to a certain extent during the expansion of the steam. This is usually such a small amount, however, that it is carried off with the exhaust. In starting an engine, care should be taken to admit steam slowly, or steam will condense too rapidly to be carried off through the exhaust pipe. With jacketed engines this condensation is much less.

In some cases the opening of the exhaust pipe is thoughtlessly placed beneath the surface of the water in the condenser, or too close to it, as shown in Fig. 4. Then, when the load on the engine is suddenly dropped, the governor cuts off the steam supply and a vacuum is formed in the cylinder. A typical indicator card of the case in point is shown in Fig. 5. If the exhaust discharge is located at *C*, Fig. 4, there will be a rush of water to satisfy this vacuum when the exhaust is opened. For this reason care must be taken to see that the exhaust opening is a sufficient distance from the surface of the water.

S. R. TYMSTRA.

Detroit, Mich.

Riding Cut-off Valves

In the August issue, page 485, Mr. Wakeman in an excellent article on Engineering Blunders says that in order to make the slide-valve engine more efficient "somebody invented another valve to 'go

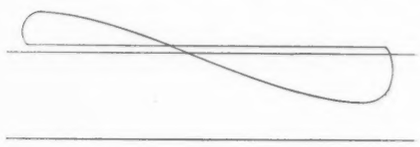


FIG. 5

on the top of the main slide-valve.'" Every constructor I ever knew gave it the name of its German inventor, Meyer, and even this writer does not know the date thereof, only that it was before 1840; which decade (from that date to 1850) was the most prolific of those steam improvements now rated so highly. In 1840, a Jerseyman invented and patented a valve that gave a 1¼-inch port to admit boiler steam; on the reverse motion, the port opening changed to 2½ inches to release the exhaust steam at rapid motion. In 1842, the first compound engine was patented by a Swede, named Carlsund. The high- and low-pressure cylinders were of equal bore and stroke, yet the low-pressure cylinder was three times the capacity of the high. The dash-pot cut-off was also patented in the same year.

I think Mr. Wakeman should have explained the use of the wheel (or handle, as it was often made) outside the steam chest; that there are two valves on the top (or side) of each D valve, which are made to approach or recede from each other by means of right- and left-

hand threads cut on the valve-stem which is revolved by the outside handle, and has a valve-stem motion that reverse (nearly) of that of the D valve beneath. With this construction, the cut-off is varied by turning the handle outside the chest, but on several engines built by the writer, the wheel had teeth cut on it, and a gear-slide was used on one, and a geared sector on others, which were linked to the governor-stem; so that the motion of the governor rotated the valve-stem; so producing an automatic cut-off.

Jefferson, Iowa. PETER VAN BROCK.

"Piping"

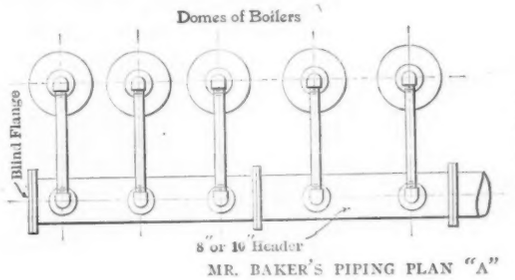
On page 573 of September POWER, "Pipe" shows the machinist's and the engineer's idea of connecting up five boilers, and in both cases I can see no provision for the expansion of the header.

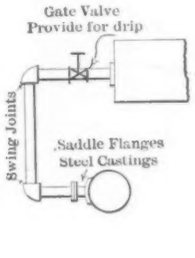
Inclosed herewith are two sketches designated "Plan A" and "Plan B." Not knowing the conditions, the size of the header cannot be figured, but I believe an 8-inch or 10-inch pipe will be large enough.

In plan A the 3-inch piping to the header has two ells, thus making a swing joint at each ell. The header is made of two lengths of pipe, with five saddle flanges, steel castings, riveted or bolted to it. Making the header all of one size of pipe allows for future steam lines to be taken from it, which usually happens in any plant. Expansion must always be provided for, or leaky joints will be the result.

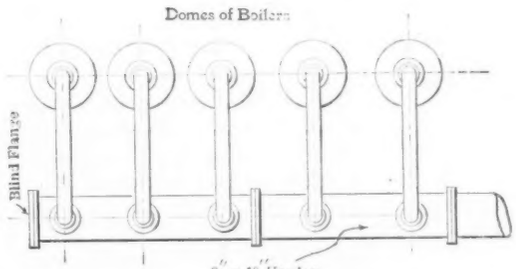
Plan B is similar to Plan A but each riser is made of one piece of 3-inch pipe bent to a U with about a 15-inch radius. This will also take care of expansion, and makes fewer joints; it is considered good practice for large steam piping. The header is the same as that shown in Plan A, with flange joints in all cases, which I prefer to screw flanges. In designing a header, always provide for expansion; cut out all sharp bends, using long sweep bends instead, have as few joints as possible, and provide for thorough drainage at every place where water can collect.

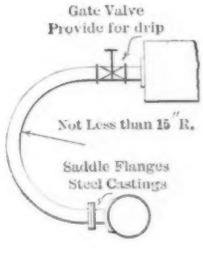
Allentown, Pa. JOHN I. BAKER.





MR. BAKER'S PIPING PLAN "A"





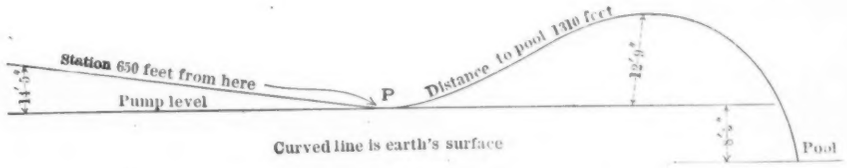
MR. BAKER'S PIPING PLAN "B"

Wanted: Advice on Pump Location

As a member of the POWER family, I will ask some other member to help me solve a problem about which I am very much undecided. I inclose a rough drawing, which I hope is plain enough to be understood. We are operating an ice plant in connection with our light plant, and have tried pumping water by gasolene engine, but find it very unsatisfactory; and as the distance is considerable to carry steam from the power house, we thought of locating a steam pump at *P* on the drawing. We have submitted this to two prominent pump manufacturers, and get unsatisfactory reports. One says it will work and one says it will not. We want about 50 or 60 gallons per minute. At the crest of the hill we can lower the suction pipe from 6 to 8 feet for any distance necessary.

If this layout is practicable, what size pump and pipe would be advisable? Can we pipe steam from the power house to the pool without too much loss?

Lawrenceburg, Ky. JOHN B. ASH.



PROFILE OF SUCTION AND DELIVERY LINE

Sand in Pump Suction—Belts

Referring to Mr. Bedford's remedy for Mr. Janicke's pump trouble as illustrated on page 701 of the November issue, I will say that it is *not* a sure cure for sand under some conditions. I had the 8-inch suction pipe of a condenser pump piped that way at a plant located in South Carolina, where because of the sand top surface on a clay bottom throughout that section, we were troubled in stormy weather with sand floatage. I had no check valve in the pipe, as it was located in the forebay and the pump had only about four feet of lift; otherwise the arrangement was the same as Mr. Bedford's.

After a time we began to be troubled with a scarcity of water, and drawing off the water from the canal one Sunday disclosed this bend in the pipe almost entirely filled with sand. Under conditions where the sand is in a quiescent state, this scheme would probably work all right, but like many other things around the power plant, it must conform with the local conditions.

On page 695 Mr. Strohm revives the old question about the belt running to the "high side" of the pulley, but the "dispute of long standing," to which he refers, will be brought to an abrupt termination, if the parties thereto will erect two shafts as much out of parallel as those shown in the sketch and try to run a belt on the pulleys of these. They will find that the belt will run off at the point *I* in the sketch, or on the "low" side.

I have had a belt trouble in practice that I have never seen discussed, and if any of your readers can prescribe a remedy, I, for one, would be glad to know about it. Here is the problem: Locate two shafts eight feet center to center, one above the other and 10 degrees out of the vertical plane. Both shafts to be level and as nearly parallel as it is possible to get them; let the top shaft be the driver, running at 228 revolutions per minute, and on this shaft place a pulley 48 inches in diameter and 10¼ inches on the face. On the lower or driven shaft place a 24x10¼-inch friction pulley that transmits the power through gearing on one end of the shaft to a heavy machine. Belt up these pulleys, using a heavy double leather belt and consider the direction of rotation such that the belt requires to be a cross belt. When the belt is idle, it will run in the center of both pulleys, but

as soon as the load is thrown on, the tight or pulling side of the belt will stiffen up, and the slack side, which is then as much slacker as the pulling side is tighter, will be forced off the side of the driven pulley a varying amount, depending on the stiffness of the belt and other conditions, but approximately one to two inches, so that the total belt surface on the driven pulley is equal only to an eight- or nine-inch belt. If the belt is crossed the other way it will be forced off on the other side of the pulley an equal amount.

Under these conditions what would be the best way to remedy this trouble?

L. L. ARNOLD.

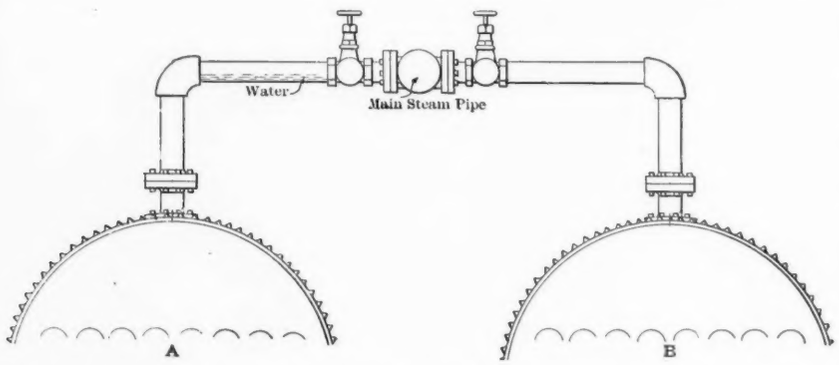
East Greenwich, R. I.

Cutting in a Boiler

Here is a trick about cutting in a boiler that everyone isn't wise to. Let *A* and *B* represent two boilers; *B* is supplying steam to the line; *A* is about to be cut in, and the pressure in it is nearly up to the required amount.

Now it is possible that some water may have collected in the pipe leading from boiler *A*. If there is enough of it, the engine might be wrecked when the stop-valve is opened. The secret of preventing trouble consists in opening the stop-valve when the steam is about five pounds lower in boiler *A* than it is in the main line; then any water that may have collected in the pipe will be forced back into boiler *A*; but be sure and open the stop-valve very slowly, and be sure that you know how nearly correct the steam gages are.

If the piping has been done correctly, of course there will be no chance for any trouble, as no bags or pockets would be allowed in a job put up by a decent steam fitter; but I have seen some pretty poor work done in piping steam plants, so it is just as well to be on your guard.



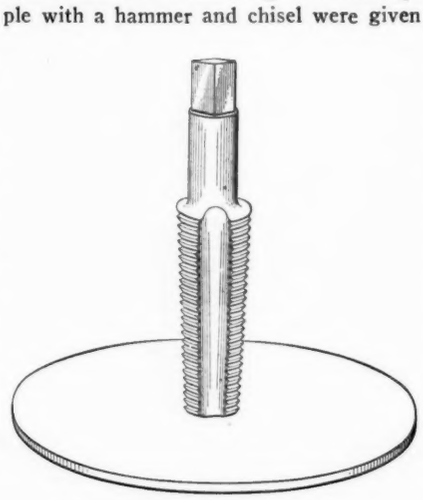
SHOWING A SOURCE OF TROUBLE WHEN CUTTING IN A BOILER

Also, in cutting in a boiler where there is a condensing engine, be careful that all the air is blown out of the boiler first, otherwise it would plug up the condenser and the vacuum would be lost.

Allston, Mass. EUGENE L. GRIGGS.

Tapping Under Difficulties

Some time ago a half-inch pipe connecting a lubricator to a pump was accidentally broken off close to the steam chest. The throttle valve leaked a little, just enough to keep a spatter of hot water coming from the hole. After the usual methods of removing a broken nipple with a hammer and chisel were given up, we rigged up a long drill in a brace and drilled through without difficulty. The tap was short and brought the operator in range of the hot water. There was not room to use a tap wrench, so I cut a piece of one-eighth-inch gasket rubber about four inches in diameter and fitted it on the point of the tap. It fitted tight on the thread and into the grooves, deflecting the water very nicely and the hole was easily tapped out.

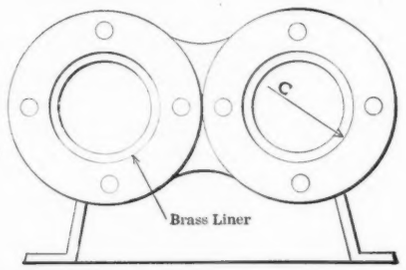


TAPPING UNDER DIFFICULTIES

La Salle, Ill. A. G. KNIGHT.

Curing a Troublesome Pump

A good-sized duplex Worthington boiler-feed pump having a lift of about 18 feet and a 4- or 5-inch suction pipe about 300 or 400 feet long gave trouble by persistently losing its suction. They called me in to see if I could better the matter, and having found the head, plungers and both valves tight I had the head taken off of one of the water ends, and the opposite end of the cylinder filled with water. No water came through the plunger, but a small stream flowed through under the brass liner which had been put in a short time previous. It developed that the old liner had worn out entirely and allowed the plungers to cut the bottom of the cylinder as shown, and that enough air worked around from the delivery to the suction side of the pump to spoil the vacuum and prevent it getting water. A good job of calking on both ends restored the pump to the normal working condition.



CURING A TROUBLESOME PUMP

Augusta, Ga. C. R. MCGAHEY.

A Problem in Transformer Connections

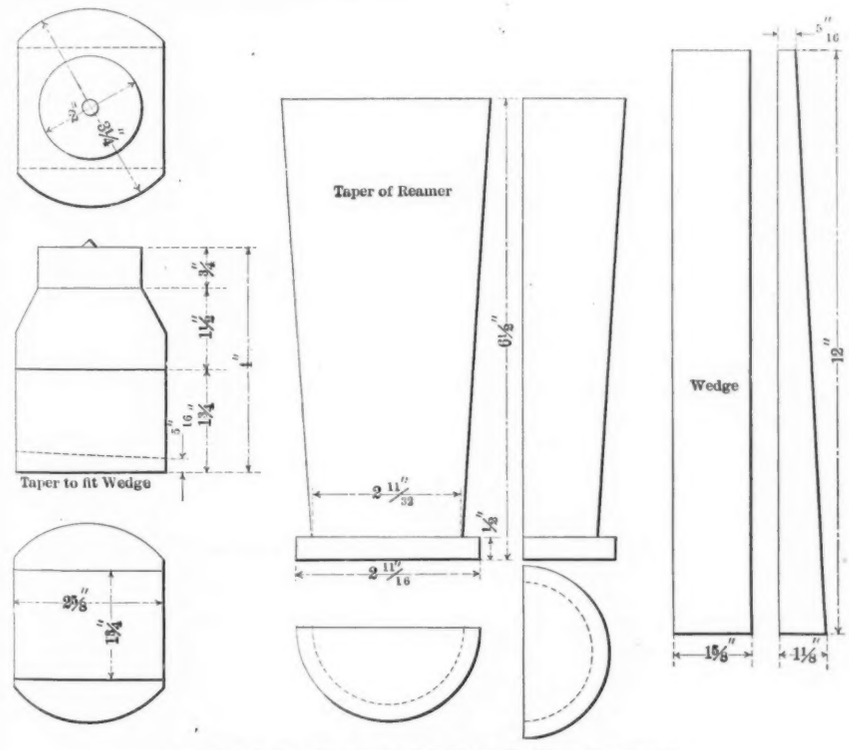
It is desired to transform three-phase currents from a high voltage to two lower voltages; one of the lower voltages is to be double that of the other. To be more specific, the primary electromotive force is 2080 volts and one of the secondary voltages is to be 110, while the other is 220 volts. Three transformers, exactly alike, are employed for the purpose. Each transformer has four coils, two for the primary, and two for the secondary. The primary coils are designed for 1040 volts *each*, while the secondary coils are designed for 110 volts each. The problem is to connect the primary coils so that they can be connected to the three-phase mains either delta or star fashion. The primary voltage will always be 2080 volts. The secondary coils of *each* transformer are to be so arranged that they may be connected in either series or parallel, and all three transformers connected in delta or star. The right instantaneous polarity

must exist between the two coils of each transformer, so that all may be delivering energy to the secondary system. In other words, the primaries of the transformers are to be connected to the 2080-volt three-phase mains in either delta or star and the secondaries are to be so connected that they can be operated upon the delta or the star connection and deliver either 110 volts or 220 volts to the three-phase secondary mains. No more than four switches are to be used, and all must be of ordinary commercial type. Four-pole switches are not to be considered. The switches must be so arranged that a short-circuit will be impossible, and no electrical connection must exist between the primary and secondary circuits. The secondary terminals of *each* transformer must always deliver the same voltage as those of each of the other two.

Boston, Mass. E. S. LINCOLN.

Piston-rod Drift

Inclosed I send blueprint of a tool for drifting piston-rods out of crossheads which is used in this section on locomotive repair work. It is more effectual for that purpose than any other tool I ever saw. You are at liberty to publish it. It was designed by John Best.



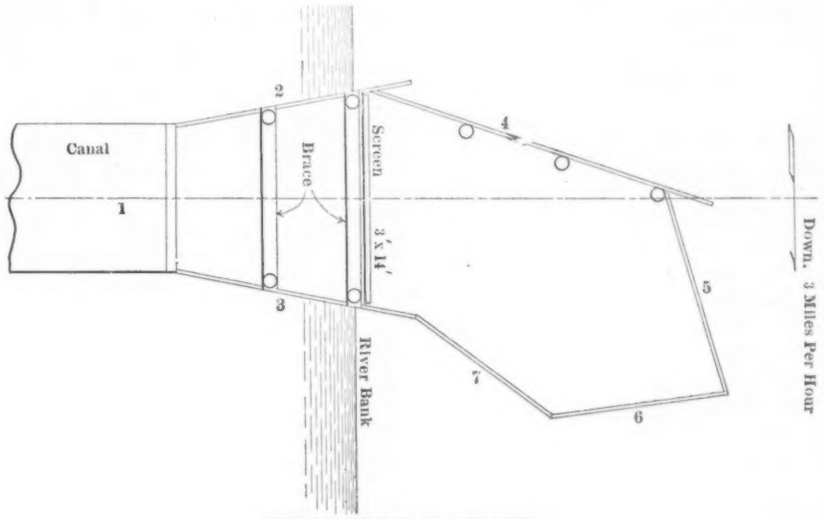
DRIFT FOR REMOVING PISTON-RODS FROM CROSSHEADS

Athens, Pa. JOHN F. KINGSLEY.

The Screening of Intakes

The men who work in places where they take their water from city mains know but little of the bad half hours that a suction from a river can give a man, especially at this time of year, when the leaves are falling. The river here is about 100 feet wide, 8 feet deep and has a current of about 3 miles per hour. From this we pump about 6000 gallons per minute for condensing and other purposes. The suction of the pumps is supplied by a canal 10 feet wide and 6 feet deep, as shown at 1 in the sketch of intake. There is about 3 feet of water in the canal at the present time. From the mouth of the canal wings were built outward, as shown at 2 and 3. Good, stout posts were driven down into the dirt and planked up on the outside, then braced across the top so the weight of the sides would not shove them in. Then a screen was made from screen wire with ¾-inch mesh fastened on a frame of 2x4 pieces 14 feet long.

This screen is set vertically in front of the outer posts and brace and can be lifted out to be cleaned. Then another wing, 4, was built diagonally out into the river 24 feet, which gave a depth of water of 8 feet. Three posts were driven, then planked up on the outside from as low down in the water as a man could work up to as high as the top of the canal. After this, old lumber was used to sheet this wing up vertically outside of the plank. The boards were driven in the mud and nailed at the top to the plank. This gave a tight wing clear to the bottom. Next two 2x12 planks 14 feet long were fastened together edgewise by spiking two pieces of 2x6 across them. In the same manner two more sections were made of 2x12 plank 12 feet long. These were all fastened together with large strap hinges and arranged as shown at 5, 6 and 7, being turned on edge and tied to the wings with ropes and held in about the position shown by other ropes. This arrangement made a very good skimmer, and when the floats had been weighted down into the water, so that only about 3 inches stuck out, they still left 6 feet of water under the outer one and quite a good depth under the other two.



THE SCREENING OF INTAKES

In ordinary conditions the leaves and weeds are all floating. They come down and strike against wing 4 and have to move out to deep water. After that 5 keeps them from going in, unless they are badly water-logged; then they will be drawn under the plank and be lodged on the screen farther up. Compared to the number of leaves floating, we get very few on the screen.

From our experience it would be a good plan to make No. 5 in the shape of an L, leaving the lower part of the L to project out; then the leaves could not be drawn under so easily. In operation the water-logged leaves come under No. 5 about two-thirds of its length from No. 4; then they follow back along No. 5 to No. 4, then follow No. 4 up to the screen. This

intake has been very cheaply built and has saved a great deal of time taking trash out of the pumps and condensers. No doubt when the spring freshets come, No. 4 will be washed away, but 5, 6 and 7 can be drawn out on the bank of the river and saved. We use a garden rake to clean the leaves off the screen and look after that work two or three times per day.

Riverton, Ill. J. W. M.

A Smoke Story

I mixed business with my vacation last summer, and spent a week or so at a time in Cleveland, Pittsburg and Wheeling. Of course I got interested in the chief town topic during my stay in each city. For instance, in Pittsburg, the several newspapers were kicking at the smoke that was being needlessly made, and were howling for laws to prevent the nuisance. A week later I was in Cleveland, where they have all kinds of laws to prohibit any kind of smoke-making, but the Cleveland smoke is quite as rich as that in Pittsburg, to which is added the stickiness and stench of crude oil.

It so happened, on a day favorable to producing smoke in large quantities, that I was in Cleveland in the locality of Broadway and the Erie railroad, where enough crude oil arose from the tanks to mingle with the smoke for a while, then the compound fell in chunks big enough to be felt, the only means of recognition through the murky atmosphere, except by sight at very close range.

I got permission at the office of a popular steam plant to interview their firemen, and without any unnecessary ceremony, I said to the first one whom I met:

"It's half past ten, and I don't see why you have the lights still burning. If it is not daylight now, the prospects are that you will have to wait till tomorrow for it. By the way, I see that since the passage by the council of the last ordinance, and the appointment of fifteen more inspectors—"

"An ye'r one of the new batch o' young fellers that's drawin a fat pay fer pokin ye'r nose into others' business. Do ye think I'm makin smoke fer the fun of it? It was only yesterday that one of the old crowd was here, an I told him, and ye can smoke it in ye'r pipe too, 'If ye want smokeless firin, gist bring on ye'r steam less ingin.' Then the chump begin to talk about the British terminal unicks that was going up the stack doin no good to anybody, but making everybody swear-in mad. Then he says to me, 'Gist let some wind in above the grates, an ye won't make any smoke, anyhow, not much.' An I says to him, 'An ye'll see the pinter on the fog clock goin back from the 150 mark without bein told, if I do as ye want me to, but I'll blow for the Boss, an when he comes, yees can do as yees like!'

"When the manager come in, he seen at onct what the matter was an he says to the party that was lookin wise: 'No, ye can't do nothin o' the kind. We'r law abidin O. K., when min make the laws, but when idiots makes' em, it's different. Jist tell ye'r folks we'll fight. Good mornin.'"

Of course, the man behind the shovel took my remark more seriously than I had intended he should. The object of my call was only to learn the steaming value of the fuel he was using, from the fireman's standpoint.

In order to get some reliable information in regard to the smoke problem, I interviewed every "Cop" that I found in the smoke belt in Cleveland, with the following result: Ten per cent. would not express an opinion, especially to a nonresident. Fifty per cent. thought the volume of smoke greater, since it was "abated" and forty per cent. of them could see no change in regard to the matter during the last few years. Several had made arrests when urged to do so. None of them complained of the smoke being unhealthy, dirty or even unpleasant, and it is a policeman's duty to be on to every phase of the public's interests.

Youngstown, O. Amos Price, Jr.

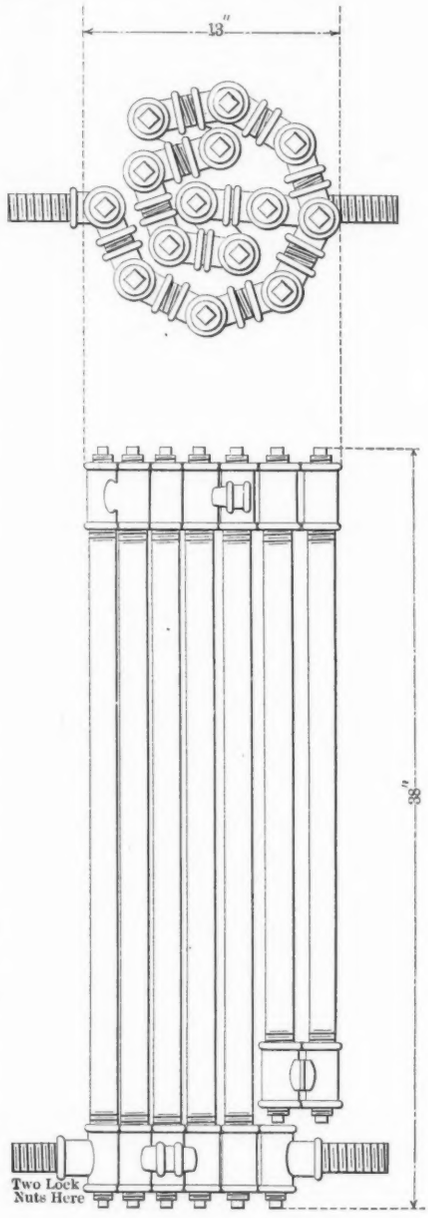
A Get-at-able Coil Heater

In our section of the country the water usually contains quite a large amount of lime and it is quite a problem to keep a closed heater in operative condition. One plant in particular here is (or was) equipped with a feed-water heater composed of a double coil of copper pipe set vertically inside a cast-iron case. This piece of apparatus has given considerable trouble owing to the coil plugging up with lime. We have so far been unable to find any way of effectively removing the deposit and so have been compelled to discard the coil whenever it got too full of lime to pass the required amount of water.

Besides the bother and time required to change the coils, they are quite expensive, so we have determined to, if possible, construct a coil which may be readily cleaned and have made one like that shown in the accompanying drawing. This coil is made of ordinary pipe fittings and pipe, and, as will be seen by the drawing, is provided with clean-out plugs at each end of every run of pipe; this leaves only the short connections between the pipe tees without straight clean-out holes, but as these are so short they may be easily cleaned from the regular clean-out holes with a bent tool.

The drawing will, I think, show with sufficient clearness the construction of the coil without much explanation. The coil is made up of close nipples, malleable tees, pipe plugs and pieces of pipe cut to the required length, and should be commenced at the center, working toward the outside with each length; the threads are all right-handed.

The two long nipples at the inlet and discharge ends have a long thread in order that lock-nuts may be used inside and outside to pack the joints with the casing.



A GET-AT-ABLE COIL HEATER

The clean-out plugs should be put in with graphite pipe compound in order that they may readily start when required.

Edw. Kilburn.

Spring Valley, Minn.

When the Feed Gives Out

There is nothing which annoys an engineer, old or young, so much or adds to his stock of gray hair any faster than a balky pump or injector. A man can usually worry along with a hot journal, or a pound which threatens to wreck the

engine, until quitting time, but trouble with the water supply calls for prompt attention and a careful diagnosis.

The writer has had the usual allotment of trouble with leaky suction pipes, clogged strainers, dirty injectors, hot feed-water. leaky check valves, broken pump valves, air-tight steel supply tank and the like, but nothing serious until taking a vacation from the home plant he went out and took charge of a traction engine just for pastime. The engine was equipped with a crosshead pump and injector. The pump was not altogether reliable, the plunger being badly worn and the injector could not be persuaded to work in any manner, shape or form. The usual routine of cleaning, examinations and experiments failed to help the matter or reveal the cause of the trouble, until a visiting engineer who had been invited to examine it found the delivery tube out of line. On questioning the owner it was learned that the injector had once been frozen. The crosshead pump often in need of repacking was beguiled into doing its duty until quitting time by putting a few drops of oil on the rod.

However, the other fellow's experience is always the more interesting. Experience is said to be the best of teachers but with even such a teacher people go wrong. I know a man who could not get a pump to work and had to have an expert come out from town and turn the check valve end for end which he had put in the delivery pipe the day before. Consequently when the pump on another engine refused to work, he insisted on taking the check valve off and turning it around. The owner, however, knew a thing or two and had another engineer on the job the next morning.

My father delights in telling of a little incident which happened to a man in his employ learning to operate a traction engine. He had been with the machine two or three seasons and my father thought nothing of leaving him in charge one day when called away on business. He was barely out of sight, however, when the check valve in the delivery pipe stuck open, when Jake shut off the injector, and let the water out of the boiler through the overflow. There was a globe valve between the check and boiler but Jake never thought of it and grabbed a wrench and commenced fumbling around the injector and piping.

Nobody thought of pulling the fire.

Ike, one of the separator men, whittled a plug out of a pine stick and got busy, dodging Jake and several farmers who were standing around giving sundry bits of advice and each trying to get his hand into the game. Ike would get his plug nicely started in the overflow, getting his hands severely burned in the operation when out it would come before he could hit it with a hammer. Finally Ike, giving up in despair, yelled, "Run boys, run, she's going to blow up!" Everybody waited for someone else to make the start, and Ike yelled again, "Run I tell you, she's going to blow up!" He himself, however, never thought of running. When lo—the wrench slipped out of Jake's hands, hit the check valve, jarred it shut and the show was off. Later in the day Jake was heard to mutter to himself about a d—d f—l trying to run an engine. But today he is one of the best engineers in that part of the country.

CHAS. E. THRASHER.

Newkirk, Okla.

Leaky Pipe Joints

In reply to "Puzzled" in the December issue of POWER, page 792, I am quite certain the following will explain the cause of the leaky joints he complains of:

Judging by the size of the pipes, I presume the threads were cut in a lathe, and my experience has been that many lathe hands are very careless about having their thread-tool ground exactly to gage. This will cause the male and female threads to leave an opening as shown in Fig. 1. The imperfect matching of the threads is sure to cause the joint to leak.

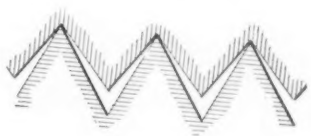


FIG. 1

Another cause of leaky joints is that the leading screw being worn causes the cutting of what is known as "bastard" threads. For example, the screw, instead of six threads, as the change gears were originally designed for, will measure five and three-quarter threads to the inch. This will cause a very imperfect fit when an imperfect tool is used for cutting the thread.

Leaky joints are also partly caused by the "spring" of the thread-tool due to wear and structural weakness of the lathe carriage and its parts. The end of the pipe in this case will have somewhat the appearance of a fluted column, indicated in Fig. 2. Of course, the sketch is slightly exaggerated to make my meaning clear.

FIG. 2

It is not an unusual thing to find the three defects mentioned in one joint Threads should be cut perfectly smooth, so as not to cause unnecessary friction when screwing up. I would advise "Puzzled" to try all the threads both on the pipe and fittings with a "thread counter," obtainable from machinists' toolmakers. The heavier the pipe and fittings, the greater the care and skill required to make a tight joint. Light-weight pipe and fittings will usually "give" enough to make up for defects in the threads.

Another very important point in securing tight joints on large pipes is to have the pipe as cold as possible and the fitting quite hot, say 350 degrees. Anyway, have it hotter than the steam that is to pass through it. It requires some little practice to know how much to allow for expansion of the different sizes of pipes. It is best to rely entirely upon the threads being in perfect contact *at all points*, to insure a tight joint, and not rely on mixtures of red and white lead, as is very often done.

Of course, some lubricant on the threads is necessary to prevent undue friction when screwing up. I have found a mixture of common varnish and graphite best. It acts both as a lubricant and cement.

I am quite certain the leaks complained of are due solely to mechanical causes—that is, imperfect threads—and not to chemicals in the water. It seems to me it would be impossible to keep tubes in the boilers if the water were so heavily charged with chemicals as to eat out the joints on the steam lines.

About the only chemical in the swamp water having an affinity for iron would be tannic acid, and while it might be present in the surface water I should hardly expect to find it in appreciable quantity at the depth of forty feet. At the depth mentioned swamp water is more likely to be charged with iron. I would ask "Puzzled" if he has trouble with the piping on which the threads were die-cut. I suppose he has noticed that the pipe does not enter the fittings far enough to "shoulder" in them.

Denver, Colo. A. H. HALE.

It is rather hard to answer "Puzzled," in the December issue without knowing more about the case; perhaps it is due to imperfect threads, expansion or contraction of the piping, or the jar caused by the machinery attached.

I am pretty sure that any acids or chemicals in the water would not affect perfect threads if they were made up tight.

A header of the size mentioned should be made up with flanged fittings and the companion flanges on the pipe should be made up tight with the pipe extending slightly through the flange.

The whole should then be faced off to insure perfect alinement. If this job is done right it would be easier for any chemical to pass through the side of the pipe than through the thread.

I was bothered with leaks in an 8-inch steam line, which led from the boiler to the engine, and after calking until I was clear out of patience, I took the line down, removed the flanges, and found the threads were wavy. After recutting the threads, making on the flanges and facing the whole off, I had no further trouble.

It is almost impossible to cut a good thread without a pipe machine, as the thread cannot be given the proper taper.

If "Puzzled" will give us a little more data and a sketch of the piping, we may be able to locate the trouble. H. G. S.

Kansas City, Mo.

Cams versus Eccentrics for Gas-engine Valve Gear—The Two-stroke Cycle Championed

The article by Mr. Junge on large gas engines in the December number of POWER is, as all his articles are, extremely valuable and instructive; but for the sake of helping to attain perfection in gas-engine design, I am taking the liberty to give my opinion on a small portion of the said article. In Fig. 105 (reproduced herewith as Fig. 1), a drawing is given showing the Klein arrangement of valve-operating mechanism for the inlet valve. I will admit this is extremely ingenious in design, but I cannot agree with the writer that it forms a cheaper construction than the old cam arrangement. Unfortunately, an eccentric is not adapted to causing a quick motion at the end of its stroke without resorting to some sort of cam arrangement between it and the valve stem; then why not use the cam arrangement directly instead of with an eccentric? The half speed of the eccentric would seem to present difficulties in the way of properly adjusting and timing at each end of the eccentric stroke, and the cheaper construction is certainly that of the cam arrangement.

The accompanying sketch, Fig. 2, will serve to show the simplicity and cheap construction of a pure cam motion, and while there is one pivoted joint more on the cam design than on the eccentric shown, it is a cheap form of joint, and if a slide was substituted for the joints 3 and 4 the two joints would be done away with.

In addition to the joints mentioned there is the double-cam lever which the roller rides on, and which is more than equal to the cost of making a cam, as the latter is an extremely cheap part to manufacture, with tools arranged for the purpose. Then there is the addition of the eccentric to make, which represents about the excess of cost of the Klein arrangement over the old-style cam device. This is not intended a criticism of design, but to show that users of the direct-cam arrangement can offer a cheaper construction and one with as long life as the eccentric style for two-stroke-cycle engines.

In reference to the undecided advantages of the two- and four-stroke cycles over each other, there appears to be a growing appreciation of the large two-stroke-cycle engine, which type should have the support of engineers and designers more than it has had. The future will no doubt witness a great many two-cycle installations. It would seem also that an ideal design would be a double cylinder vertical with the lower end of the cylinders used for compressing the charge, instead of exploding the charge at both ends for the sake of accessibility and cheapness of construction, the only difficulty being in the successful cooling of the piston.

Oshkosh, Wis. L. J. MONAHAN.

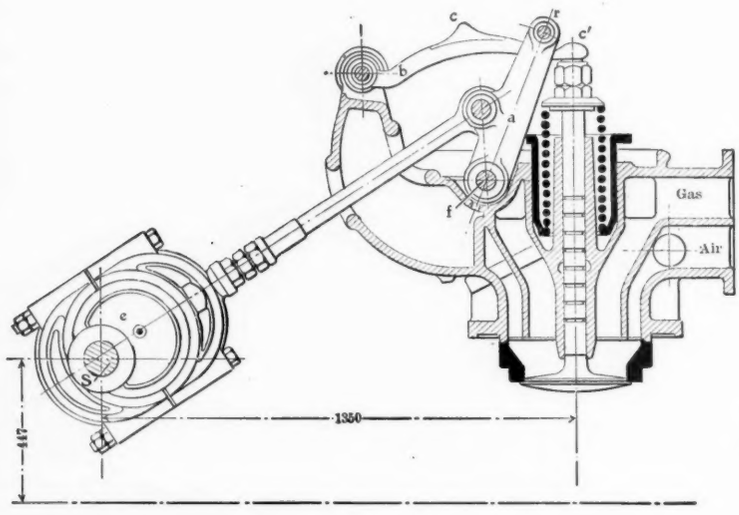


FIG. 1. (REPRODUCED) SHOWING KLEIN ARRANGEMENT OF VALVE-OPERATING MECHANISM FOR INLET VALVE

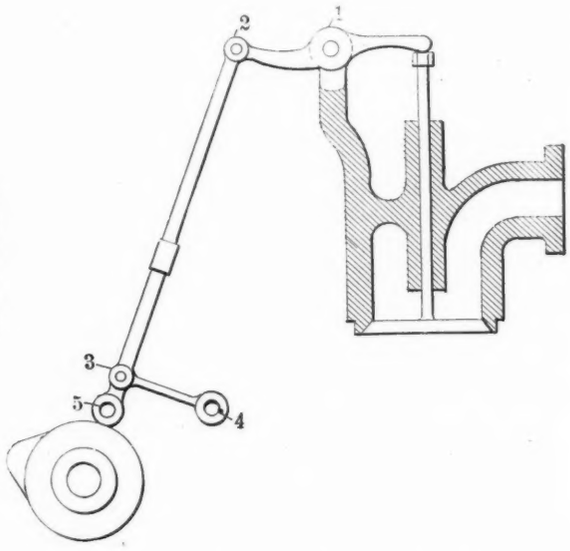


FIG. 2. SHOWING SIMPLE CONSTRUCTION OF A PURE CAM MOTION

Top Feed for Boilers

When I wrote the letter on "The Feed-water Inlet to Steam Boilers," which appeared in November POWER, I really did not think that any reader would take note of or answer it; but as a few have criticized it, I believe the "boys" have not quite grasped my idea of the "top feed," so I made the accompanying sketches to help clarify the situation. Fig. 1 depicts my idea; Fig. 2 shows the layout the critics evidently thought I meant; Fig. 3 represents a side view of one of my 60 and 16x44 four-inch tubular boilers.

A is the inlet, in front of the boiler; *B* is a brass bushing; *C* is the feed-pipe, which runs between the water level and the top row of tubes; *D* is an ell, looking and discharging the water crosswise of the boiler; *E* is a shaving and sawdust pipe.

I once had charge of a boiler in which the feed entered at the point *F*, in which case the bottom sheet and back end became covered with scale, the tubes would grow together, and the flues leaked.

I have seen the arrangement in Fig. 2, but I believe with "Student" (in December POWER) that it is not good judgment to drop feed-water on the sheets directly over the hot fire. Therefore, as shown in Fig. 1, I lengthened the pipe so the feed-water would be carried to the front end. One can readily see that by having the shaving and sawdust blowing in at *C*, the hottest part is back of the bridgewall and the coolest is at the front.

Starting at the heater at 212 degrees Fahrenheit, the path of the feed-water is through the pump, inlet *A* and pipe *C* around to *D*. By this process, if the engineer is cautious in feeding and keeps one water level, the feed-water will be nearly at the temperature of the water in the boiler, the circulation will hardly be

affected, steam will be easier, and scale can be more readily removed than by any other system using Mississippi river water.

Leroy Baker's idea, in the December issue, of having the inlet between the third and fourth rows of tubes, is possibly all right, but it looks to me no better than the bottom-feed system; and as long as the feed-pipe is under water a little kerosene or boiler compound pumped through will remove all trouble with sediment in the pipe.

A lot of arguments can be put up on the different ways of preparing feed-water, as well as the right place for it to enter the boiler. Some engineers like one way and some another. I am a firm believer in filtering the water before it enters the boiler, instead of waiting until the water has been boiled. Ordinarily exhaust steam will not heat the water to the right temperature, neither will the live-steam heater.

I tried the following plan and found that it works pretty well on a small scale: Cut out the tubes of a boiler, converting it into a plain cylinder boiler. Pipe the exhaust steam to one end of this boiler, or heater (the larger the boiler, the better), and pipe the other end for the outlet. The suction pipe to the boiler feed starts about midway between the bottom and the water level. This is a cheap heater, holds water enough to permit compound to settle the sediment, will heat the feed-water, if one water level is maintained, to about 206 degrees Fahrenheit, and will deliver the water filtered better than will many exhaust-steam open heaters now on the market.

If one had two settling tanks, each large enough to run six hours, he could let one stand six hours while using the other, or use one in the forenoon, and then fill it up; after dinner use the other, and then let them both stand with a little compound in each tank. By that means one would get pretty good water.

I followed this plan in our plant for a while, but my company wanted something "up-to-date," and they got it. If they had it to do over again, however, those tanks would be installed again. The new heater we have is a "peach" for heating up to 212 degrees Fahrenheit, but the other plan discounts it on filtering, although it loses a little in temperature. The old device was rigged up for two hundred dollars, while the new apparatus cost about three hundred fifty dollars.

My style of heater is as good an oil extractor as any, too. The oil will float on the top of the water, pass out through the overflow, and not as much oil will enter the boiler as with the up-to-date heater. I will admit that the new heater is a better ornament to the engine-room than two big settling tanks would be, but I think that if the tanks could be placed in the cellar, with the feed-pump low enough for the hot water to run into the water cylinder, the result would well pay for the bother and expense. Purer water would be obtained, and by using my plan, shown in Fig. 1, the engineer would be in clover.

Muscatine, Ia. QUINCY T. HOWARD.

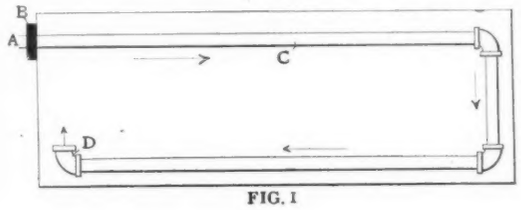


FIG. 1

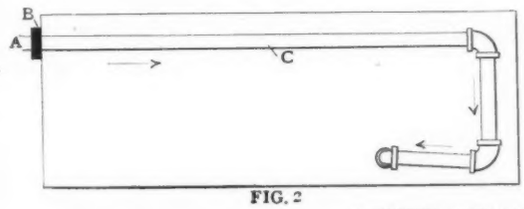


FIG. 2

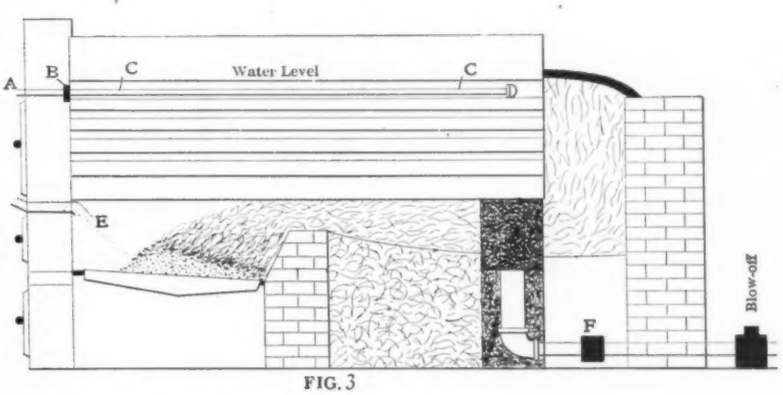


FIG. 3

SKETCHES EXPLAINING MR. HOWARD'S VIEWS ON BOILER FEEDING

Mr. Junge Replies to Critics

[On page 108 of this issue will be found the beginning of discussions by Dr. Chas. E. Lucke and R. E. Mathot, the Belgian expert, on F. E. Junge's paper on "The Evolution of Gas Power," which was read at the December meeting of the A. S. M. E. These discussions were afterward submitted to Mr. Junge, who replies to them in the following letter.—EDS.]

Dr. Lucke's discussion is noteworthy and most vital to the question. I entirely agree with him that the limitations of the gas engine and especially its relations to the steam engine have not been sufficiently emphasized in my paper. Unfortunately, that portion of the paper which was especially devoted to the question upon which he enlarges was cut out by the editing staff of the society, probably because otherwise the paper would have been abnormally long. I can also quite understand that the situation as I presented it, basing my remarks on my familiarity with European conditions, and the remarkable performances which I recorded, must seem optimistic and even impossibly high to one who had limited his energies to the study and development of the American gas engine.

In this connection, I find that the figures which our distinguished friend Monsieur Mathot submits in his discussion are very apt to convince those who may doubt the correctness of my statements, that the efficiencies recorded have actually been attained on the European continent. I believe that when commenting on the evolution of gas power it is logical, necessary, and proper to give, besides the average performances of gas prime movers, which were graphically represented by several curves, also the best figures which have so far been attained, both with large and small engines, as long as they are clearly designated as maxima.

I may add that it has given me a great deal of satisfaction to note how much more interest was revealed toward our subject at the recent meeting of the society, compared to that exhibited on previous occasions. I shall welcome any criticism or comment that may be advanced in the discussion of my paper in the columns of POWER as long as it is based on strictly scientific grounds and is dictated by the earnest desire to benefit the development of the art. F. E. JUNGE.

New York.

We are advised by H. L. Clements, secretary of the Yukon Stationary Engineers Association, at Dawson, that difficulty is experienced by engineers coming into the territory, under contract or otherwise, and ignorant of the fact that an experience of at least one year in the territory under an engineer holding a territorial license is essential to the obtaining of a license permitting one to operate a steam plant. Mr. Clements will be glad to furnish information in regard to the demand for engineers, the current rate of wages, etc., to anyone who is contemplating going into Yukon.

POWER

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Simplicity, Elaboration and Ultimate Efficiency

The young engineer learns as matters of accepted fact that a tubular boiler is more economical than a plain cylindrical one, that a Corliss or other four-valve engine is more economical than a simple slide valve, that a condensing engine is more economical than one exhausting under atmospheric or greater pressure, that a compound is more economical than a simple engine, that a heater will improve the economy one per cent. for every ten degrees, about, that the feed is heated, that mechanical conveyers handle coal more economically than laborers, that mechanical stokers are more economical than flat grates and manual stoking.

The sooner and more completely he unlearns this the sooner will he be a success and the greater success will he be.

Every item of a steam plant must be charged with the interest on its cost, the additional taxes and insurance which must be paid upon it, its depreciation in value, year by year, and the various items which go to make up the standing charges, as well as repairs, attendance and maintenance.

If one boiler costs ten dollars per horse-power and another twenty and both may be counted upon to last twenty years, the buyer of the cheaper boiler would use up fifty cents worth of boiler per year per rated horse-power and of the more expensive a dollar's worth. He would pay half the interest, taxes and insurance on the first that he would on the second, etc., and these items become an important part of the whole when the cost of a horse-power is not over fifteen dollars per year, including everything. If one had his coal given to him a Corliss engine would save him nothing over a slide valve. If he has use for much of his exhaust steam, a condenser will be a drag on the ultimate efficiency of the plant.

These thoughts are induced by the leading article in this month's issue. A plant has been put down at a ridiculously low cost per horse-power (we regret that the real figure cannot be given) partly because it was bought right, but principally because it is made up of the simplest standard materials procurable and adapted to the conditions.

Suppose this plant cost forty dollars per horse-power, buildings, foundations and everything except the land included. At twelve per cent. the standing charges would be four dollars and eighty cents per year. If its cost were more than doubled, as the preliminary estimates contemplated, the standing charges would have been nine dollars and sixty cents per horse-power.

In 1897 F. P. Sheldon collected some data regarding the cost of power in the New England textile mills, and the same year the costs of the Warren (R. I.) Manufacturing Company were published. The final figures were as follows:

	SHELDON.		WARREN.
	Lowest.	Highest.	
Fuel	4.76	6.60	4.70
Attendance	2.34	2.71	1.88
Supplies	.31	.51	.42
Repairs	.19	.31	
Fixed charges	4.04	4.72	4.55
Total	11.64	14.85	11.55

It will be seen that the fixed charges are a large proportion of the whole, approaching the fuel charges in two of the three columns. Is it conceivable that any elaboration that would have more than doubled the cost and thus the fixed charges, could have made a commensurate reduction in the fuel and operating expenses, and unless it would have shown a very considerable net saving at the end of say twenty years, would it have been good business to put so much money where it could not have been gotten out?

The department in which the plant is most simple is the boiler-room. Horizontal return tubular boilers may be considered—shall we say provincial?—but who shall gainsay their efficiency and low first cost if one has the space to use them? A couple of hundred tons of coal a day is a large amount to handle with men and shovels in these days of machinery, and good machinery, for that purpose, and Mr. Diman will consider with an open mind any proposition to attain the results which he is attaining at less expense, everything considered.

Ignition in Gas Engines

It has happened several times within our observation, and doubtless hundreds of times beyond it, that changing the ignition system on a gas or gasolene engine has resulted beneficially. In some cases the change has been from low-tension make-and-break to high-tension jump spark; in others, the reverse. This does not necessarily indicate that the one system is best adapted to certain engines and the other one to other engines; in controversion of this theory is an experience with a gasolene engine of moderate size on which the ignition system was changed first from jump spark to the make-and-break system and afterward from the latter back to the jump spark, improvement resulting from both changes! The improvement obtained by the first change was due merely to the production of a slightly "fatter" spark; in making the second change, the spark-plug location was changed, putting the spark in a more favorable position with respect to the charge in the cylinder, the original spark plug having been located in a pocket, where the fresh charge was contaminated by burned gases forced into the pocket by the compression stroke of the piston.

In the abstract, the make-and-break sys-

tem should more closely approximate certainty of ignition and result in more efficient flame propagation, because the spark produced by that system is larger and less likely to be impaired by the compression of the cylinder contents. In practice, however, if both systems are well designed and intelligently applied, there is little choice between them. It takes only an apparently trivial departure from good practice to entail unsatisfactory ignition from either system. In the case of the jump spark, an increase in the length of plug projecting into the combustion space has been known to convert a battery-demolishing and entirely unsatisfactory system into a smooth-working, economical one. The incasing in rubber tubing of the secondary wire leading from the spark coil to the spark plug has produced a similar metamorphosis, and five minutes of intelligent effort in the adjustment of the spark-coil vibrator will often work wonders.

The make-and-break system probably embodies fewer vulnerable points, but there are several features which one cannot afford to neglect or abuse. The contacts, even when they are of the wiping variety, need cleaning thoroughly at reasonable intervals, the frequency depending, of course, on the character of the service. If the spring of the oscillating electrode is too light or too heavy, good results cannot be obtained, especially at high speeds; if the electrodes are allowed to separate too far, or not far enough, this may cause uncertain ignition. The inertia of the moving electrode should be considered in adjusting the point of ignition, if this is fixed by the maker; although the trigger may snap off at the desired point when the engine is turned over slowly by hand, the point of actual snapping off when the engine is running will be later than when it is turned over by hand. For this and other good reasons it is better to provide for adjustment of the point of ignition through a considerable range, as is done with jump-spark ignition, instead of providing merely a starting and running position, as many builders do with make-and-break igniters.

No matter which system of ignition is used, it is of the utmost importance that the spark be produced at a point in the combustion space favorable to prompt ignition and speedy flame propagation. The large majority of engine designers seem to concur in the opinion that this point is as near as possible to the inlet valve, where the incoming fresh charge is practically certain to drive away all burned gases remaining from the previous combustion. If the sparking points be thus located, and if both systems be well designed and arranged so that the time of ignition can be adjusted closely to suit the working conditions, there is practically no choice between them, except for very high speeds, when the jump-spark system will nearly always give the better results.

Standard Regulations for Making Gas-engine and Producer Tests

It has been pointed out in our columns at various times by different writers that a necessity exists for some recognized standard to which tests of gas-power equipment should conform in order that such tests might afford proper basis of comparison. Now that the gas-power industry is taking such strides forward in this country, the need for such a standard is becoming acute. Not only is a recognized code of test methods and procedure needed for a rational basis of comparison between different plants or between any given plant and a standard performance, but it has become urgently necessary as a means of enabling prospective purchasers of gas-power equipment to judge as to the soundness of claims made by manufacturers' representatives for their apparatus. It is seldom that two makers follow out identical procedure in determining the output ability and the thermal efficiency of gas producers or large engines. Consequently, statements as to the performance of different apparatus cannot be intelligently compared by the layman. However, we have before us the report of a suction-producer test of just sufficient duration to cover the period of water-gas production—fuel at maximum incandescence and the maximum permissible quantity of steam going through—the result being, of course, a heat value for the gas that no suction producer could by any possibility maintain throughout a run of practical duration. Of course, this test was made innocently by the interested parties under the impression that they had a perfect right to test the producer under the most favorable conditions and claim the results to be characteristic of that producer, but if there had been in existence a recognized code for making such tests perhaps this would not have happened.

The case just mentioned is only one of several of its general class that have come up within the near past; hence we say that there is urgent need for a code of procedure for testing gas-power apparatus similar to that recognized in steam practice. In another part of this issue F. E. Junge submits what might be called the suggestive basis for such a code. It is not claimed that the rules submitted represent the finality of thought on the subject; they were taken from a code prepared by a committee of German engineers and may be considered as a foundation on which a code of regulations adapted to American practice may be formulated. We imagine that the majority of the proposed rules, just as they stand, could be adopted profitably, but some of them, of course, would require adaptation and it is probable that a few additional provisions might be found advisable. However this may be, the suggestion is timely and the nucleus furnished by Mr. Junge is ample and skilfully prepared.

Engine-stops and Fly-wheel Insurance

We deplore the controversy, indications of which appear in our correspondence columns, between the promoters of fly-wheel insurance and the manufacturers of engine-stops. One would expect to find here a complete community of interest. There is no doubt that safety stops have prevented many fly-wheel explosions, that their use has saved many lives and much property. A disastrous fly-wheel accident is heralded abroad and becomes known of all people, but the explosions which are headed off by the operation of the safety stop are seldom heard of outside of the plant or of the engine-room. The safety stop is really the safety valve of the engine. It has been compared to a secondary or auxiliary safety valve, the governor being the main safety valve, but the comparison would be more just as regards a boiler which depended upon a damper regulator to prevent excessive pressure. The addition of a safety valve to such a boiler would be more nearly representative of the addition of a safety stop to an engine. It would seem therefore that companies insuring fly-wheels would not only favor, but insist upon the use of these devices. On the other hand, the safety stop is a mechanical device, fallible as are all human devices. Boilers explode despite the use of safety valves; engines run away even when provided with safety stops. No man eschews boiler insurance because he has his boilers provided with safety valves; an engine-stop is no substitute for fly-wheel insurance. When it stops an explosion it may save lives that money could not pay for, and prevent damage from delay, loss of profits and interruption to business that is not included in an insurance policy, but if through untimely interference, as in a recent instance, or of having been put temporarily out of commission from one cause or another, the safety stop should fail, it is well to have an insurance policy in the safe to cover the material loss. The safety-stop man who tries to make a customer by telling a power user that the use of his stop will make fly-wheel insurance unnecessary is as wrong as the agent who urges the absolute protection of an insurance policy against the fallibility of the mechanical device. The owner of any engine the bursting of the fly-wheel of which is liable to create disaster should guard against such breakage by the use of an engine-stop, and then insure himself against loss from the failure of the stop, or from the breakage of the wheel from some cause other than overspeed.

The Buffalo Gas Engine

The A. H. Alberger Company, Buffalo, N. Y., is building a line of single-acting tandem gas engines which embody a number of original features. Fig. 1 herewith shows the general appearance of the single engine on the valve-gear side and Fig. 2 shows a twin engine from the other side. The prime feature of differentiation from common practice is the use of single-acting cylinders in tandem. The piston-rod from the rear cylinder passes through a stuffing-box in the head of the front cylinder, and the piston in this cylinder is of the trunk type, with the connecting-rod wrist-pin set across it. Fig. 3 is a longitudinal section of the engine and illustrates the constructional features of the cylinders, pistons, and other large parts. It will be noticed that the rear piston is made short and light; this is practicable because there is no side thrust on this piston.

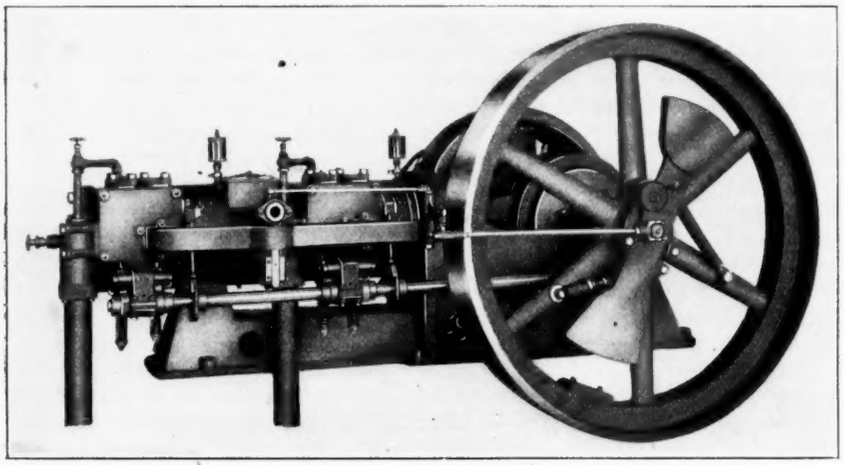


FIG. 1. BUFFALO TANDEM GAS ENGINE

Access to the rear cylinder is obtained by removing the head, which, it will be noted, is not water-jacketed; removing the connecting-rod, unbolting the piston-rod and drawing out the trunk piston gives access to the front cylinder. The removal of the connecting-rod is easier than an inspection of Fig. 3 would indicate. The wrist-pin is held in the piston by two caps which may be readily unbolted by means of a socket wrench; then the connecting-rod with the wrist-pin in its box is withdrawn. Fig. 4 illustrates this construction, which is a front view of the front piston. Fig. 5 is a view of the two pistons and the piston-rod, and shows the arrangement of packing rings and the shoe which catches oil from the front oil cup, shown in Fig. 3 and delivers it to a small pipe inside the piston which carries it to the wrist-pin oil funnel.

The piston-rod runs in a water-jacketed stuffing-box which is packed with cast-iron spring rings. These rings are similar to the ordinary cast-iron snap rings used in pistons, except that the "spring" is not outward, but inward, so that the rings clamp the piston-rod. The casing surrounding and carrying the rings is in two water-jacketed halves bolted together, the joint being ground. The flange of the complete stuffing-box is then bolted to the front cylinder head. Oil is fed between the rings, as indicated in Fig. 3. The stuffing-box is said to require no adjustment, repacking, or attention whatever. The manufacturers state that these stuffing-boxes have been in use for several years without receiving a moment's attention, and, without exception, working as perfectly as when the engines were new.

The crank-shaft and connecting-rod are made of open-hearth steel, each forged in one piece and cut from the solid. The crank is fitted with balance weights, which are secured to the crank-disks. The connecting-rod is fitted at the crank end with a bronze box of the marine type lined with babbitt metal, and with a bronze box with wedge adjustment at the piston end.

The frame is a single heavy casting of the double-wall type, strongly braced and designed for great strength and rigidity. It is sloped well forward from the main bearings, and is spread laterally for distributing the weight on the foundation. The metal is carried in straight lines between the front cylinder yoke and the main bearings. The main bearings are split at an angle, so that the rigid frame receives the pressure of the shaft when the engine is in operation. The bearings are lined with babbitt metal. An oil groove is cast around the bottom of the frame to protect the foundation. A door is provided on one side of the frame for convenience in grouting when erecting the engine on foundation. The interior of the frame is used as a settling chamber for any grit or foreign material which may be present in the air used by the engine. The door on the side of the frame has holes through which air is admitted, and the air passes to the intake pipe from a point near the rear end of the frame extension which supports the cylinders. The cylinders are cast separately and bolted together and to the frame, this method of construction being considered as insuring positive alinement.

The usual secondary side shaft imparts motion to the valves and igniters. It is driven from the main shaft through two-to-one spiral gears, the engine operating on the Otto or four-stroke cycle. It may be incidentally mentioned that these are the only gears used on the engine. The two valve chests and their auxiliary parts are similar in all respects. An extension from the lower side of each valve chest forms a hanger which carries a journal box for the side shaft, as shown in Fig. 1. A hand-hole over each valve affords access thereto, and also permits the valve to be withdrawn from the chest. The valves are vertical and of the poppet type with double guides. Each valve is mechanically actuated by a simple cam and lever movement. The lever is forked at the valve-stem end and pushes, when the valve is being lifted, against a collar on the valve stem. Side thrust is obviated by the interposition of a metal roller between each fork of the lever and the lift collar on the valve stem, as shown in Figs. 6 and 9. It will be noticed that this

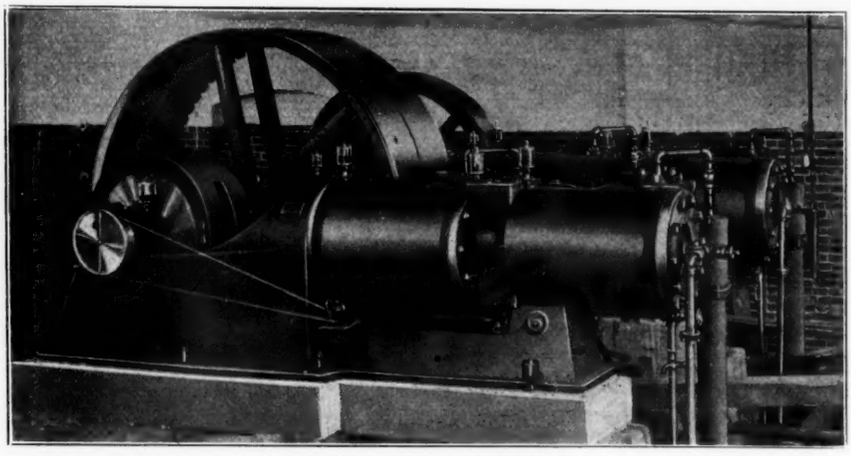


FIG. 2. BUFFALO TWIN TANDEM GAS ENGINE

roller is flattened on top, affording a sliding contact with the lift collar, so that it rotates only as much as may be required in its cylindrical bearings in the fork of the lever. This arrangement is said to prevent side thrust entirely. The tension of the valve spring is adjusted by means of the nuts shown on the lower end of the valve stem. The valves are made with square bosses on the heads to permit regrinding in position. The two valve-stem guides are cast-iron bushings, which are easily and cheaply renewable. Each valve is lifted from its seat by the lever and closed by its own weight assisted by the spring.

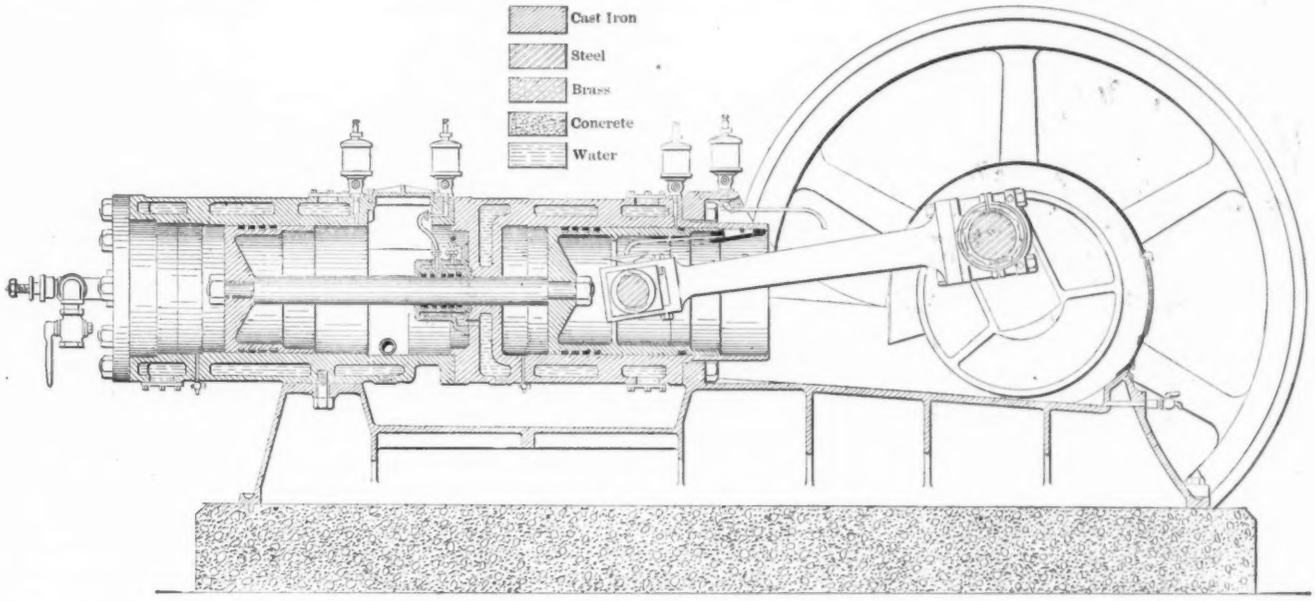


FIG. 3. SECTION THROUGH CYLINDERS AND PISTONS OF BUFFALO GAS ENGINE

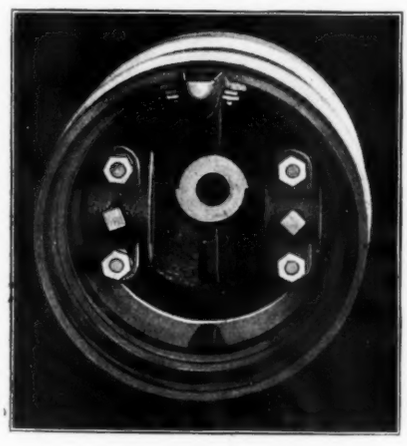


FIG. 4. FORWARD PISTON

The igniters are of the electric make-and-break type, mounted on the sides of the valve chests. The igniter electrodes are mounted in a plug which can be easily removed and replaced; the plug has a ground joint which requires no packing. The igniter points are located in the path of the cool entering charges over the admission valve and away from the exhaust valve. The hot exhaust gases, therefore, do not pass over them, and the fuel mixture which surrounds them at the instant of ignition is uncontaminated by the burned gases remaining in the clearance space. The breaking of the electric circuit is caused by a trip, one of the electrodes in the plug being rocked by an eccentric on the side shaft, as illustrated in Fig. 7. The time at which ignition occurs may be adjusted while the engine is in operation. The point at which ignition is occurring is indicated by a small pointer on an exposed dial. Before starting the engine, the igniters are set at a late ignition point, and when it has reached normal speed the igniters are set at the proper running point. The adjustment is made by means of quadrants, one of which is shown in Fig. 7. A rocking cam is oscillated by the eccentric on the cam-shaft, and its nose engages with the igniter trigger on the downward motion, carrying the end of the trigger down until the circular path of the cam nose is separated from that of the trigger far enough to disengage them. Then the trigger is snapped back by a spring, separating the igniter contacts in the valve chest. The rocking cam is pivoted on an eccentric stud, the end of which is squared and fits in a hole in the end of the quadrant lever; rocking the quadrant twists the stud in its socket, and thereby moves the center of the cam pivot toward or away from the igniter trigger, thus altering the point at which the cam and trigger become disengaged.

Speed regulation is effected by means of an ingenious mixing valve controlled by a Rites inertia governor, mounted on the fly-wheel as in steam-engine practice. The governor and its linkage are shown in Fig. 1, and the mixing valve is illustrated by Fig. 8. The valve is cylindrical, with a partition across the middle of it separating the ends into two chambers. Ports are cut in the walls and the valve sets in a cage having similar ports cut in its wall. The valve and cage are shown separately at the central right-hand part of the drawing. The valve cage is mounted in a cross pipe bolted to the valve chests; this pipe comprises the mixing chamber and carries the fuel mixture to each cylinder. The valve is free to rotate in the cage under the control of the governor. Gas is admitted to the interior of the upper end of the valve, and air is admitted to the interior of the bottom end; the gas and air pass through the valve ports to the mixing chamber and thence to either cylinder.

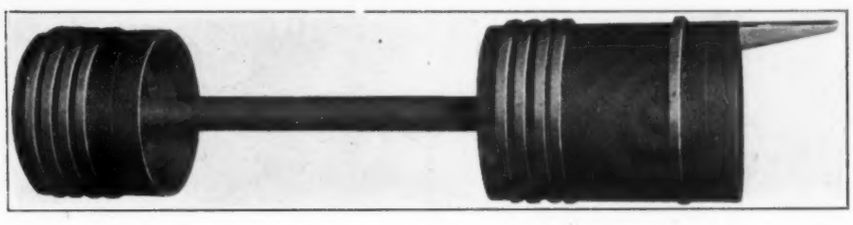


FIG. 5. PISTONS AND PISTON-ROD OF BUFFALO GAS ENGINE

At the beginning of a suction stroke, the governor partially rotates the valve so that its ports come opposite the cage ports, and gas and air pass through the valve and cage to the cylinder. A reverse movement caused by the governor closes the valve earlier or later during the suction stroke, cutting off simultaneously both the gas and air supply from the cylinder, the valve remaining closed during non-suction strokes. The valve is bal-

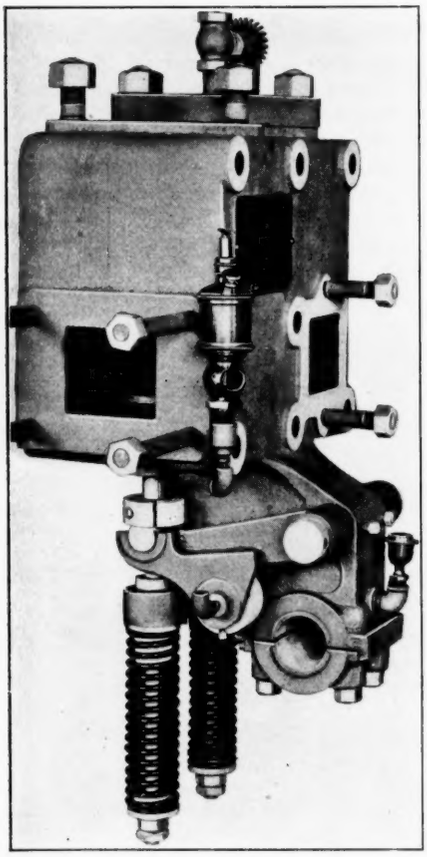


FIG. 6. SHOWING VALVE-LIFTING LEVER

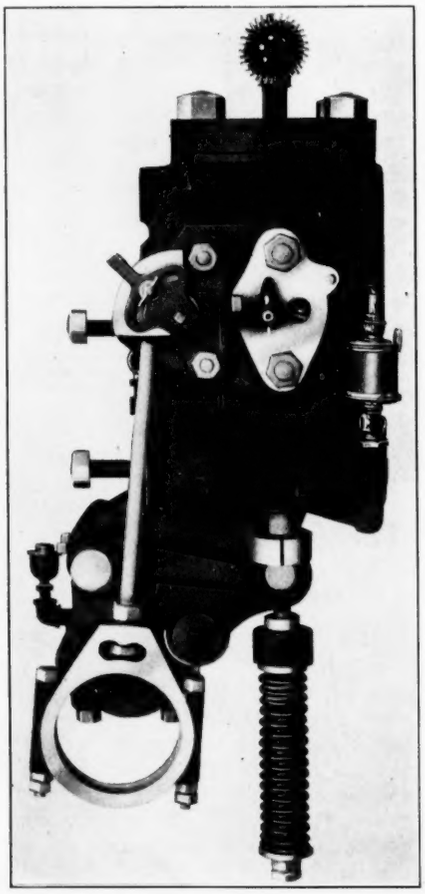


FIG. 7. IGNITER MECHANISM

anced, of course, and the multiple ports are said to insure thorough mixing of gas and air, and to obviate the wire-drawing that would occur with an ordinary throttling valve. Moreover, there is no possibility of the gas flowing into the air inlet during non-suction strokes; and when suction producer gas is used and the gas is below atmospheric pressure, the air cannot flow into the gas inlet during non-suction strokes.

This valve is also adjustable to secure the relative proportions of gas and air required by the quality of gas used. It is arranged so that it may be moved up or down by hand; if moved upward, the effective area of the gas ports is reduced and the effective area of the air ports is increased. If moved downward, the reverse effects are obtained. The movement of the valve upward or downward is accomplished by turning a thumbscrew on the exterior of the mixing chamber. This thumbscrew has a fine thread, so that the movement of the valve may not only be carried through a wide range, but may be varied by micrometer gradients to secure the exact proportions of gas and air. Since movement of the valve up or down increases the area of one set of ports by exactly the same amount that the area of the other set is decreased, the total effective port area remains unchanged, no matter what the adjustment of gas and air proportions.

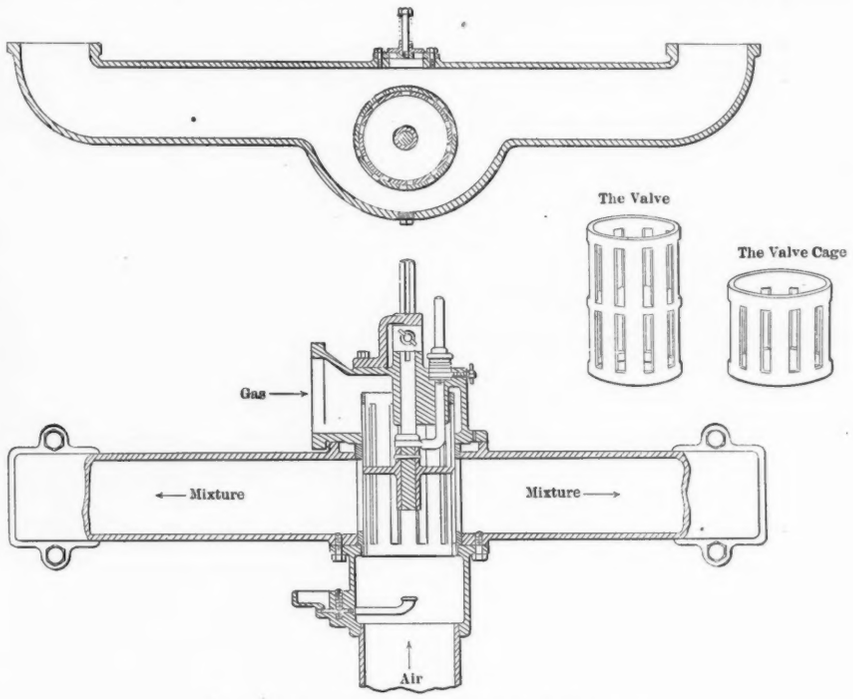


FIG. 8. ALBERGER MIXING AND REGULATING VALVE

Since the two pistons move forward and backward together, the power stroke of one occurs simultaneously with the suction stroke of the other and the compression stroke of one occurs with the exhaust stroke of the other, the cycles in the two cylinders being "staggered." This obviously gives the crank one impulse per revolution, as in a single-acting single-cylinder steam engine, and permits the travel of the governor to be appropriate for the intake requirements of both cylinders without recourse to extra mechanism for changing the relation between the governor travel and the valve motion of one cylinder.

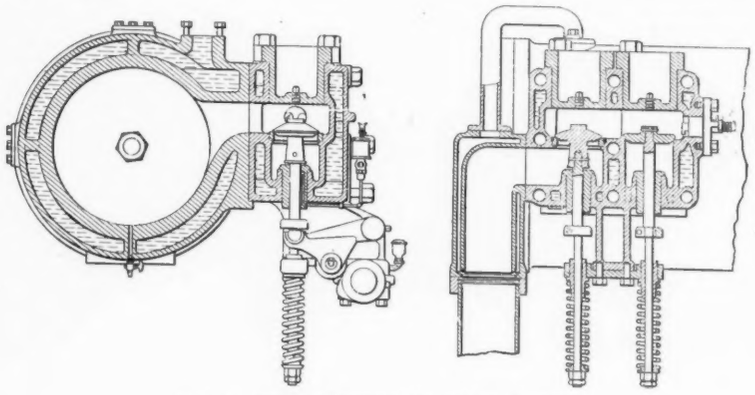


FIG. 9. CROSS-SECTIONS OF VALVE CHEST

The jacketing of the engine is clearly

shown by Figs. 3 and 9, and the latter also shows the arrangement of the valve chamber. The jacket water, after passing through the jackets, is mixed with the exhaust gases immediately outside the valve chamber, thereby reducing their volume and temperature and relieving the back pressure on the engine. This is accomplished in mixers, attached one to each valve chest on the end opposite the igniter. As this mixing and cooling is accomplished at the engine, there are no excessively hot pipes or parts where the operator might burn himself. The portion of the jacket discharge water evaporated by the exhaust gases passes out with the gases as steam, and this exhaust instead of giving a loud report is quiet and resembles the exhaust from a steam engine. The exhaust passes through a centrifugal separator in the exhaust pipe, where the unevaporated waste jacket water is separated and entrained. This separator is illustrated by Fig. 10.

Lubrication is effected by simple means.

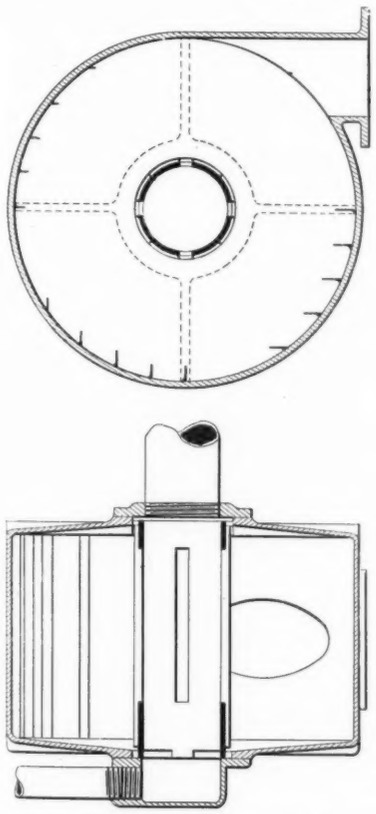


FIG. 10. EXHAUST SEPARATOR

The main bearings are self-oiling, with oil rings on the crank-shaft to prevent oil from creeping along the shaft onto the wheels or the armature, in case the engine is direct-connected to a generator. Oil for the crank-pin is thrown by centrifugal force through ducts in the crank-disks to the crank-pin. This feed may be varied by hand and the crank-pin flooded if desired. The lubrication of each cylinder is effected by means of an oil cup on top of the cylinder near the front end, remote from the combustion chamber.

NEW THINGS FOR THE POWER PLANT

The Ehrhart Steam Turbine

In turbines of the Parsons type the thrust of the steam has to be balanced. This is usually effected by the use of balancing disks of the same diameters as the several steps of the rotors. Large units are sometimes divided, steam entering at the center and working in opposite directions upon similar turbines, the thrusts balancing each other. The objection to this is that in order that the entry area for the steam at the high-pressure end may be small enough the initial blades must be small and short, and the necessary clearance is a large proportion of the blade area, aggravating leakage losses. When the turbine is cut in two as in the above described arrangement, this disadvantage is increased.

Several modifications of the Parsons type have been made by Parsons himself and by Westinghouse, in the direction of introducing the steam in an expanding nozzle, as in the De Laval and Curtis turbines, allowing this to impact upon a central set of blades, and then proceed at the greater volume into the combined impact and reaction vanes of a true Parsons. A patent for such a turbine has recently been issued to Raymond Nelson Ehrhart, of Pittsburg.

Referring to the illustrations the steam is expanded in nozzles 1, the energy due to such expansion being absorbed in a single row of impulse blades 2. The steam, on leaving the impulse blades, collects in a transforming receiving chamber 3. The inlet 4 of the receiving chamber is provided with guide vanes 5, which prevent rotation of the steam issuing from the impulse blades. The steam passes from the transforming chamber 3 to the exhaust chamber 8, through a final section of alternating annular rows of stationary guide vanes 6 and moving blades 7, whereby the steam is fractionally expanded down to the exhaust pressure, the expansion occurring both between the rows of vanes and between the blades.

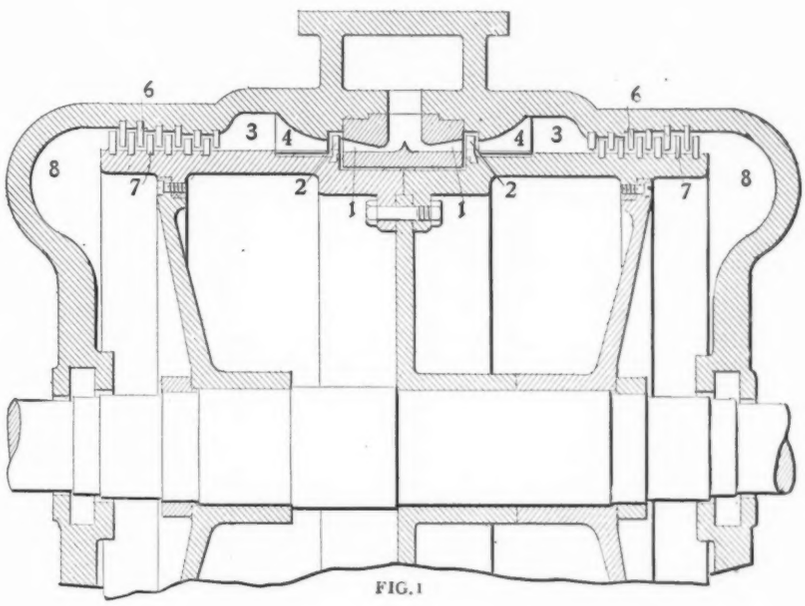


FIG. 1

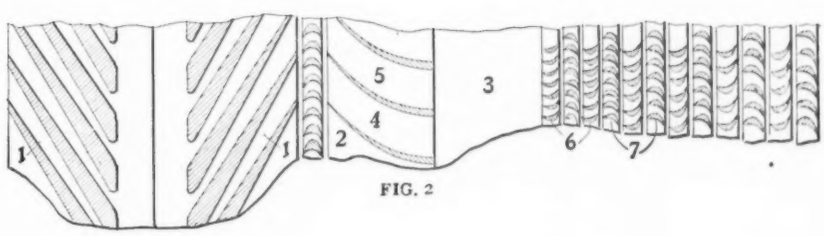
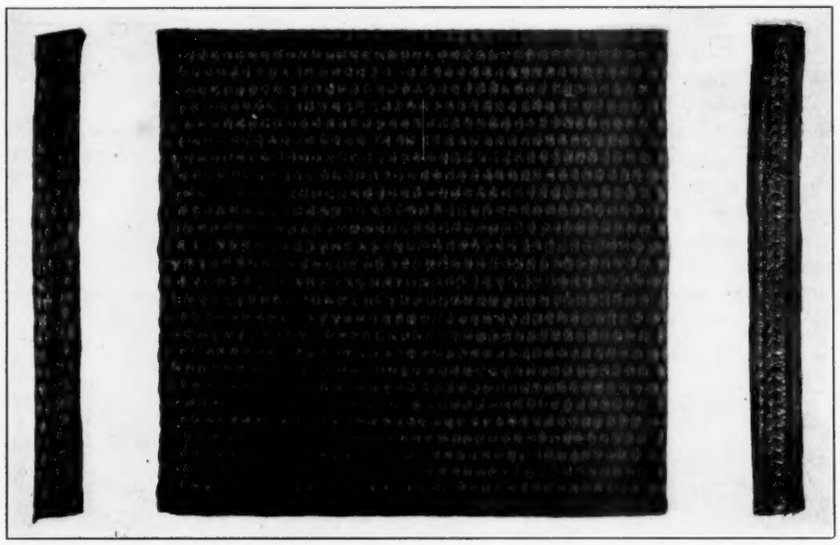


FIG. 2

DIAGRAMS ILLUSTRATING THE PRINCIPLE OF EHRHART'S STEAM TURBINE

Scandinavia MCM Belting

This is a solid-woven cotton belt, the weave of which extends through the entire thickness so that it is not divided into plies, and the selvage is complete, as shown in the accompanying engraving. The MCM denotes 1900 in Roman numerals. After being woven the belt is impregnated with a lubricating and moisture-resisting composition, giving it pliability and adhesion to the pulley.

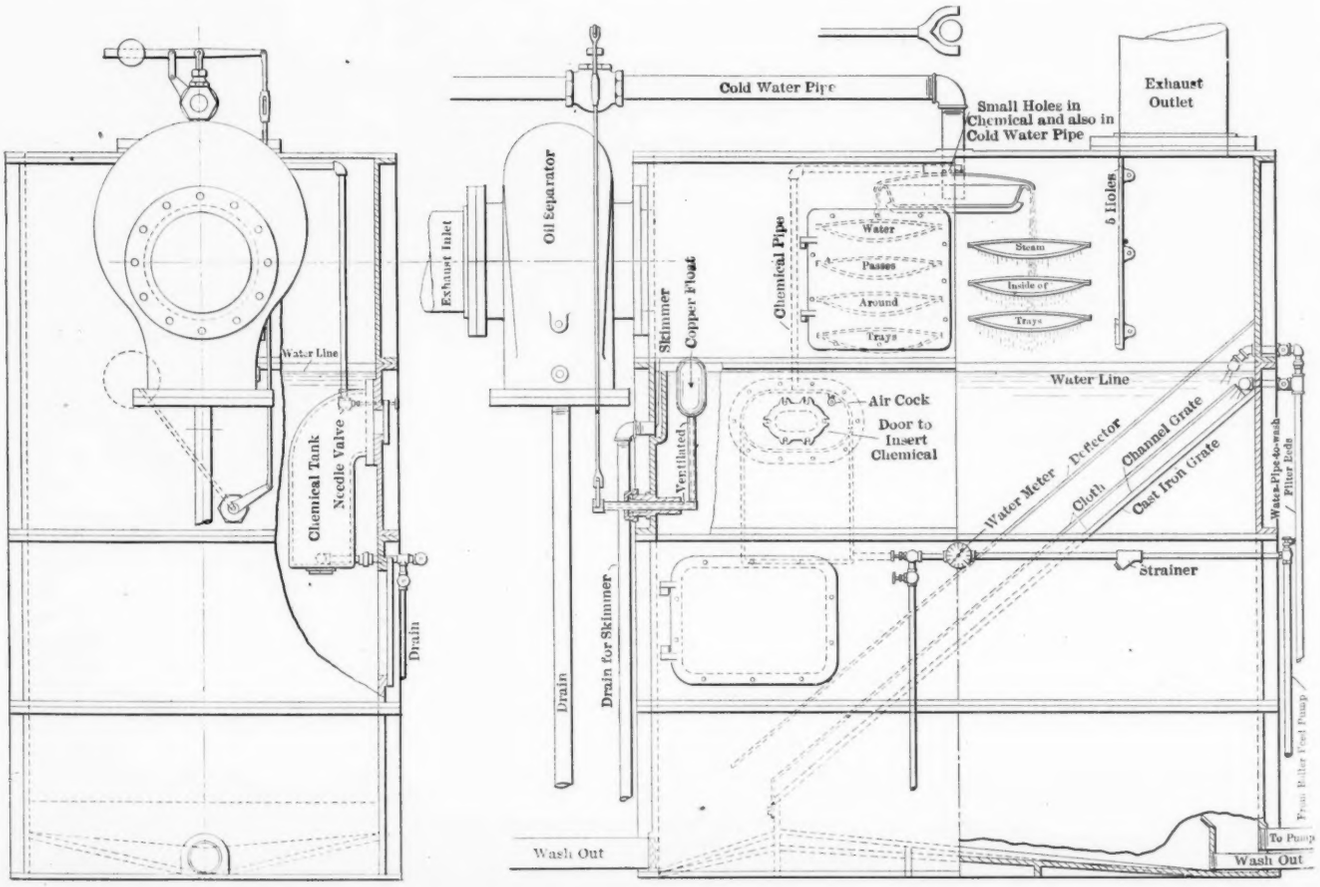


SCANDANAVIA MCM BELTING

It is made regularly in single, extra stout, and triple weights, the thicknesses being three-, five- and seven-sixteenths, respectively. Above these weights it is made up to three-quarters of an inch in thickness, to order. The single belting will sustain a stress of 1300 pounds per square inch of width, the extra stout 2300, and the triple 3000, and any of these grades are procurable in widths up to 54 inches. It is used not only for power transmission in everything from governor belts to main drives, but for belt conveyers, and is furnished by the Scandinavian Belting Company, 145 High street, Boston, Mass.

Marion System of Feed-water Heating and Purification

Referring to the accompanying engraving, the exhaust steam enters at the left and passes around and through the hollow trays and out by the curtain or deflector at the right, the deflector being provided with inclined grooves or troughs to arrest any entrained water. The cold-water supply is controlled by the copper float shown, ventilated to the atmosphere so that it cannot fill by percolation or leakage. The entering water trickles over the hollow trays, and being surrounded by the exhaust steam, as well as in contact with the metal heated thereby, is rapidly brought to the temperature of the exhaust steam and falls into the bottom of the heater, together with such of the exhaust steam as has been condensed. A chemical tank, the plate giving access to which is shown at the right of the float, and which is also shown in side view in the end elevation at the left, is filled with the reagent necessary to throw down the impurities with which the particular feed-water dealt with is charged. This tank is piped to the delivery side of the pump, as shown, and under the pressure thus exerted the chem-

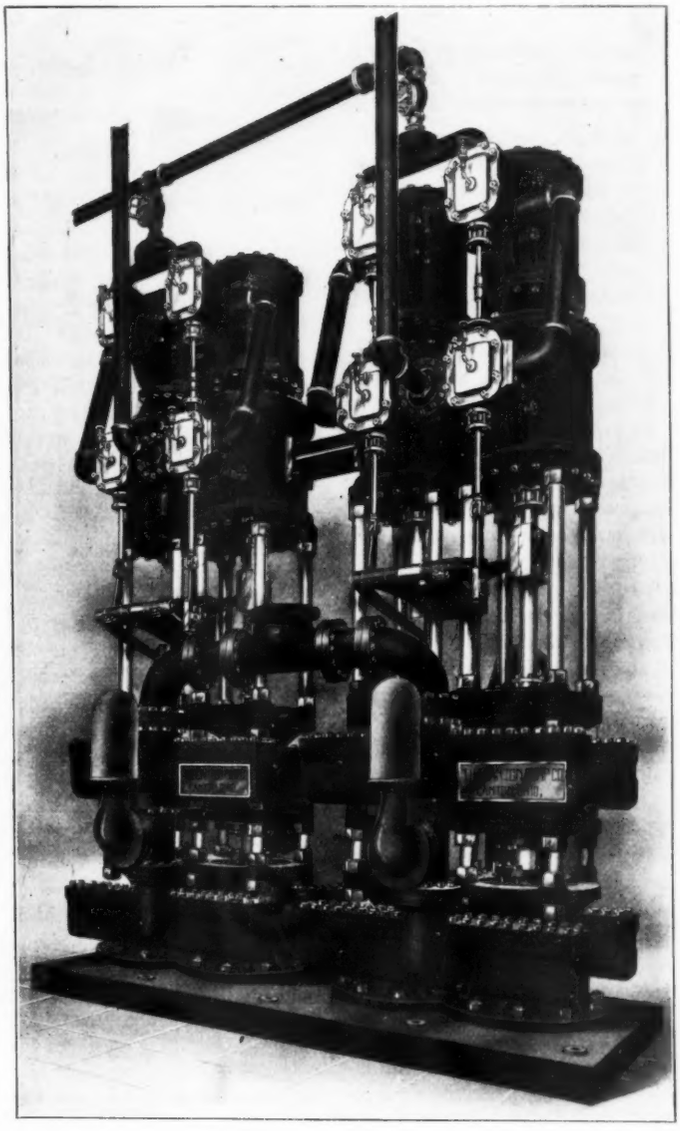


THE MARION IMPROVED FEED-WATER HEATING AND PURIFYING SYSTEM

ical is forced through the needle valve and the pipe leading to the point of entry of the cold feed-water. The delivery end of the chemical pipe is bifurcated, straddling the cold-water pipe, and both pipes being filled with small perforations, the chemical is intimately mingled with the incoming water, its action being accelerated by the heating process which the mixture undergoes immediately thereafter. In the storage space afforded by the bottom portion of the heater, the separation and precipitation of the impurities takes place, the heavier ingredients settling, to be discharged through the wash-out pipes, and the lighter or floating impurities being drawn off through the skimmer at the left of the copper float. Between the raw-water and pure-water sides of the apparatus is an engineer's inclined filter, described in our issue of March, 1905. The apparatus is manufactured by the Marion Inclined Filter and Heater Company, Marion, Ohio.

Canton Vertical Compound Elevator Pump

The *raison d' etre* for the skyscraper is expensive land. The thing that makes the skyscraper practicable is the power elevator. If plan or floor space is so valuable, it is advisable to economize it by placing the machinery necessary to operate the elevators in the vertical rather than in the horizontal position.



THE CANTON VERTICAL DUPLEX ELEVATOR PUMP

Such an arrangement of the pump built by the Canton Hughes Pump Company, of Canton, O., for the Wells building, Milwaukee, is shown herewith. These pumps have 18-inch high-pressure cylinders, 20-inch low-pressure cylinders, 6-inch plungers and a 12-inch stroke, and each pump has a capacity of 250 gallons per minute, working against a head of 900 to 1000 pounds per square inch with 120 pounds suction and 120 pounds steam pressure at the throttle. They are of the ordinary duplex compound type, but are believed by the manufacturers to be the largest pumps of this type that have ever been arranged in the vertical position. The base of the pump is in one solid casting and weighs nearly five tons. Upon this are mounted two complete twin pumps, identical in construction and either pump can be operated independently of the other. The weight of the steam end is 14 tons and of the water end and stretchers about 14 tons more. The pump is capable of exerting a pressure of 1400 pounds to the square inch.

New Varieties of Goetze's Corrugated Gasket

The Goetze gasket, previously described in several forms, is now procurable as well in the forms shown in the accompanying engravings. That shown in Fig. 1 consists of a thin corrugated copper gasket, the corrugations of which are filled with asbestos. This gives it the yielding quality required to allow the gasket to accommodate itself to uneven surfaces where a plain copper gasket would simply flatten out, while the copper gives the necessary tensile strength to prevent blowing out, and is prevented from flattening by the packing in its concentric ridges. The packing in this form is adapted to high

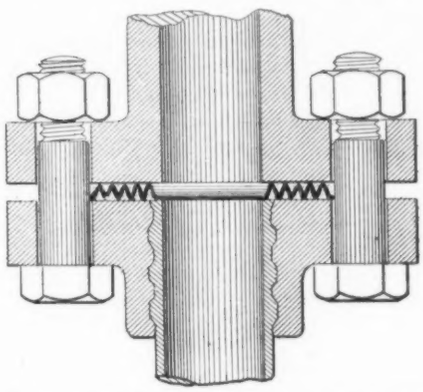


FIG. 1. GOETZE'S CORRUGATED COPPER GASKET WITH ASBESTOS LINING

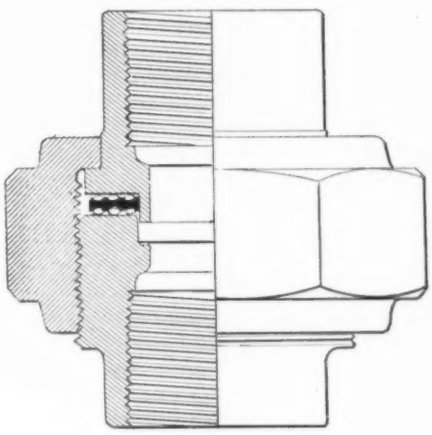


FIG. 2. GOETZE'S SEAMLESS COPPER GASKET WITH GRAPHITE FOR UNIONS

degrees of temperature and to high pressures, and remains tight under considerable variations in both. It is procurable in any breadth of rim desired and in all special shapes up to 7 feet in diameter, the width of the grooves being from 5/64 to 7/16.

Fig. 2 shows a solid seamless copper gasket which is lined with a preparation of graphite and especially adapted for use in unions. It is impervious to oil and will resist any temperature and pressure met with in practice.

"Goetzerit" is a sheet packing, procurable either in a sheet form or already cut to gasket. It is in color like a light printer's board, and a sheet 40x40 inches and 1/32 of an inch in thickness weighs 3 pounds 5 ounces. Gaskets are carried in stock from 1 to 24 inches. It is recommended for high pressure and superheated steam, and for use with ammonia, oils, air, gas, acids and alkaline fluids.

These devices are made by Frederick Goetze, 725 East One Hundred and Forty-first street, New York.

The Building of the Engineering Societies—Opening Exercises—A. S. M. E. Meetings

The first use of the building presented to the engineering societies of America by Andrew Carnegie was on Tuesday evening, January 8, when Frederick P. Fish, a member of the Institute of Electrical Engineers, addressed a meeting of the American Society of Mechanical Engineers upon a topic of general industrial interest: "The Ethics of Trade Secrets." All of the holding societies have now moved in, but the formal opening of the building will not occur until April 16, when the donor can be present.

The dedication exercises will cover two days, in fact, April 16-17, and will include also a grand session of the three societies which make their home in the building. The principal addresses on the opening days will be made by Brig.-Gen. Wm. Crozier and Brig.-Gen. Arthur Murray, who will talk on "Coast Defenses." It is thought likely that they will speak at separate sessions.

It is expected also that the usual monthly meeting of the A. S. M. E., which would naturally occur on the second Tuesday (April 9), will be postponed until the opening date, April 16.

On February 12 two papers will be presented at the A. S. M. E. meeting. C. E. Sargent, of Chicago, will read his paper on "The Testing of Inflammatory Gases," which was published in the December issue of the Proceedings. Professor Chas. M. Allen, of the Worcester Polytechnic Institute, will present a paper, or rather will lecture on "Gasolene." The lecture will be illustrated by experiments and will be followed by discussion.

On March 12, John W. Lieb, Jr., vice-president of the A. S. M. E., will deliver an address on "Vesuvius and Pompeii." It is anticipated that this will be especially interesting, as Mr. Lieb has been to considerable expense to prepare slides and data. Among other views Vesuvius will be shown in eruption.

Personal

E. L. Broome, steam engineer, of the New York Central & Hudson River Railroad Company, has resigned to join the forces of The Stone & Webster Engineering Corporation, of Boston, Mass. Mr. Broome will be succeeded by W. C. Miller, Jr., formerly assistant steam engineer.

Asa M. Mattice, formerly connected with the engineering departments of the Westinghouse and Allis-Chalmers Companies, and more recently practising as a consulting engineer in New York, has undertaken the management of the works of the Walworth Manufacturing Company at South Boston, Massachusetts.

F. N. Bushnell has resigned his connection with the Rhode Island Company and the Naragansett Electric Lighting Company, in which position for several years he had charge of the steam and mechanical engineering ends of the trolley and lighting operations of Providence and vicinity, and has accepted a position with the Stone & Webster Engineering Corporation, of Boston.

J. H. Smith, formerly business manager of *The Electric Club Journal*, Pittsburg, Penn., has purchased the capital stock, property and good will of the Electrical Age Company, which publishes *The Electrical Age*, of New York City. The offices of the company have been removed from the Cassier publication offices, at 3 West Twenty-ninth street, to 45 East Forty-second street, New York.

H. J. Lamborn, a native of Philadelphia, and a mechanical engineer and technical expert of high standing, has been appointed superintendent of power and plant of the Yale & Towne Manufacturing Company's works, to succeed F. A. Waldron, who resigned a year ago. S. E. Dauchy, who has been acting superintendent, will continue as assistant superintendent, which position he has held several years.

A banquet was given to Joseph H. Nichols, Past Examiner of Engineers, and Henry G. Lamy, Examiner of Engineers, by the engineers of Buffalo, on Tuesday evening, January 15. The mayor, the U. S. local inspector and two of the other past examiners were guests of honor. Herbert E. Stone was toastmaster, and the speeches ran largely toward the consideration of the relations of the engineer, the public and the State.

Wm. M. Davis, of Pittsburg, Penn., has accepted the position of chief of the department of lubrication for Arthur D. Little, chemical expert and engineer at 93 Broad street, Boston. Mr. Davis, through his contributions to POWER and his published writings upon this subject, is well known as an expert in lubrication, and will carry to his new connection a wealth of experience as well as the result of years of careful and analytical study of the subject.

Obituary

On January 8 announcement was made of the death of James Clayton, of Brooklyn, one of the pioneer manufacturers of air compressors in this country. Mr. Clayton was born in Homer, England, fifty-seven years ago. He began the manufacture of air compressors between the years 1864 and 1867, and Clayton apparatus has attained world-wide prominence.

The author of a paper read before the Manchester, England, Association of Engineers states that the credit of bringing the centrifugal pump on to a working basis is usually given to Appold, but long before his time the centrifugal fan was at work, for as far back as 1713 Papin designed one, and others have been made, both in England and America.

It is announced that the Yale Scientific School has inaugurated a series of courses for workingmen, either free or for payment of a nominal fee. The courses will comprise twenty exercises of four hours a week in the evening on steam and the steam engine, practical electricity, mechanical drawing and machine designing. In the courses both employers of labor and the trades' councils will be asked to cooperate.

Electric Traction Weekly is the title of the newest street and interurban railway publication. It is published at Cleveland, O., by the Kenfield-Fairchild Publishing Company, of which H. J. Kenfield is president and general manager. Mr. Kenfield was fourteen years vice-president and Eastern manager of the old *Street Railway Review*. The editor of *Electric Traction Weekly* is C. B. Fairchild, Jr. (vice-president of the new company), who for several years was associate editor of the *Street Railway Review*, and latterly associate editor of the *Street Railway Journal*. The new weekly, now about two months old, gives unmistakable promise of being, as it claims, "a record of practice and progress for men who are busy." Certainly its progenitors have a wide circle of acquaintance and a large measure of success is prophesied for it. It is of standard size, newsy and attractively got up.

The *Electric Railway Review*, formerly *Street Railway Review*, is now issued as a weekly, it having heretofore been published monthly.

Inquiries

Questions are not answered unless they are of general interest and are accompanied by the name and address of the inquirer.

Cooling Water Required by Jet Condenser

An engine using a jet condenser develops 90 indicated horse-power and uses 28 pounds of steam per horse-power-hour. The terminal pressure in the cylinder is three pounds absolute and the temperature of the mingled steam and water is 125 degrees Fahrenheit. How much cooling water is required if its temperature on entering the condenser is 60 degrees Fahrenheit? R. W. C.

There are in a pound of steam at three pounds absolute pressure 1125.1 B.t.u. above 32 degrees Fahrenheit. Supposing the exhaust were all steam of this pressure, the engine would void:

$$90 \times 28 \times 1125.1 = 2{,}835{,}252$$

B.t.u. per hour.

To raise a pound of water from 60 to 125 degrees requires:

$$93.17 - 28.01 = 65.16 \text{ B.t.u.}$$

To absorb the heat voided therefore would require:

$$\frac{90 \times 28 \times 1125.1}{65.16} = 43{,}512$$

pounds of water.

Some of the heat entering the engine in the steam is converted into work, however, and the steam giving up that heat is condensed. The more correct method would be to deduct from the heat entering the engine the heat equivalent of the work developed and the amounts taken out by the jackets if such are used, and the amount lost by radiation if you want to be very particular, and then divide the remainder by the heat absorbed by each pound of water. For example, suppose the steam came to the engine at 100 pounds pressure absolute; one pound of such steam contains 1181.8 B.t.u. The engine would therefore receive:

$$90 \times 28 \times 1181.8 = 2{,}978{,}136$$

B.t.u. per hour.

One horse-power is: $33{,}000 \times 60 = 1{,}980{,}000$ foot-pounds per hour; and 1 B.t.u. is equivalent to 778 foot-pounds. The engine would therefore convert into work:

$$\frac{90 \times 1{,}980{,}000}{778} = 229{,}048.8$$

B.t.u. per hour. Subtracting this from the above, the engine is found to void:

$$2{,}978{,}136 - 229{,}048.8 = 2{,}749{,}087.2$$

B.t.u. per hour, less the radiation and jackets. Divide this by 65.16 and you will get:

$$\frac{2{,}749{,}087.2}{65.16} = 42{,}189.7$$

pounds of water per hour.

Cost of Pumping Water

What is the cost of pumping 62,000 gallons of water into a tank against an 80-foot head, with a duplex steam pump 8x5x10 inches? The boiler shows an evaporation of 8.5 pounds of water per pound of coal and coal costs $5.70 per ton of 2000 pounds. S. F. Cottrell.

Taking the weight of a gallon at 8⅓ pounds, the work performed will be:

$$62{,}000 \times 8\tfrac{1}{3} \times 80 = 41{,}333{,}333$$

foot-pounds.

You do not give the steam pressure nor the duty the pump is rated at, nor say whether the pump runs condensing. Suppose it does 10,000 foot-pounds per pound of steam, which would be something less than 200 pounds of steam per horse-power-hour, it would take 4133 pounds of steam. It will probably take less, but it may take even more. If the boiler evaporates 8.5 pounds of water with a pound of coal it will take:

$$4133 \div 8.5 = 486$$

pounds of coal, which will cost:

$$\frac{486}{2000} \times 5.70 = \$1.38.$$

Size of Boiler Feed-water Main

How should the size of the feed-water main be determined for a boiler plant, the horse-power of which is known? C. E. J.

It depends on what you mean by "horse-power," and also on the pressure at which this horse-power is developed. The quantity of water evaporated per unit of time is the fundamental factor in the problem. If you know how many pounds of water will be used per hour by the whole plant, divide this number by 60 in order to get the cubic feet of water per hour. Divide the cubic feet by 12,000 (which is a suitable velocity in feet per hour) and the result will be the proper area of the feed-water main in square feet. Multiply this by 144 and you will have the internal area of the pipe in square inches. From a table of pipe areas you can take out the corresponding diameter. This may be condensed into a formula, as follows:

$$\frac{\text{Pounds per hour}}{5000} = \text{area of pipe in square inches.}$$

For example, suppose your boiler plant is to evaporate 75,000 pounds of water per hour. Divide this by 5000 and the result is 15, which is the proper internal area of your feed-water main. Reference to a table of pipe dimensions will show you that 4½-inch pipe has an internal area of 15.96 square inches, so this would be the size to use.

Cleaning Out an Ammonia System

I have in my charge a "Ball" ice-making plant. I wish to withdraw the ammonia, repair the plant and test the system with air pressure. Will it be safe to blow out the coils with steam and then pump, say, 300 pounds air pressure on them? The joints are all rubber packed. Would you advise repacking all joints or only those that prove leaky during test? R. H. Roark.

In order to clean out and repair the plant you should first to pump out the system well and put all the ammonia into the storage tank. During the whole time of pumping out keep plenty of water running over the condensers. Then for taking the ammonia out of the system you have to attach an ammonia drum to the point where the ammonia is taken into the system. This drum ought to be slanting backward and water, as cold as possible, should be kept flowing over the ammonia drum. By this means you might be able to get about 80 pounds of ammonia into a 110-pound drum. When you cannot get any more ammonia into the drum you have to lead the rest into a vessel filled with water so that the water absorbs the ammonia. This ammonia you have to lose; or you could send it to a factory and have the water taken from it, which would hardly pay.

Testing with 300 pounds air pressure is done in a new plant, but when a plant has been running for some time there is oil in the system, and this together with the heated air forms a very explosive gas which has caused many violent explosions. For this reason the air pressure should not be allowed in old plants. It would not be advisable to clean the coils out with steam as the steam condenses and fills the pockets with water and it is very hard to get out. After you have taken the ammonia out of the system you should open the coils at all the lower points and let the oil and dirt run out. should renew all the rubber joints on the high-pressure side, that is where the ammonia is hot, and of course all leaky ones On the suction side the rubber seldom gives trouble.

Testing Furnace Temperature

To find the temperature of a furnace, a copper ball weighing three pounds is put in the furnace and, after being heated to the same temperature, is placed in a vessel containing 35 pounds of water at 44 degrees Fahrenheit. The water temperature is raised to 65 degrees Fahrenheit; what is the temperature of the furnace? R. W. C.

To raise a pound of water from 44 degrees to 65 degrees Fahrenheit requires

$$33.01 - 12 = 21.01 \text{ B.t.u.,}$$

33.01 being the number of British thermal units in a pound of water at 65 degrees and 12 the number in the same quantity at 44 degrees above those in a pound at the freezing point.

To heat 35 pounds of water through this range would require, therefore,

$$21.01 \times 35 = 735.35 \text{ B.t.u.,}$$

and this is the heat given up by the ball.

To heat a pound of copper one degree requires 0.0951 of a heat unit. To heat three pounds one degree would require

$$3 \times 0.0951 = 0.2853 \text{ B.t.u.}$$

The 735.35 given up by the water would therefore heat the ball to

$$735.35 \div 0.2853 = 2573.9 \text{ degrees,}$$

which, plus the 65 degrees present temperature, must have been its temperature when immersed in the water.

Book Reviews

"The Energy Chart, Practical Applications to Reciprocating Steam Engines." By Captain H. Riall Sankey. Albert Frost & Sons, Rugby; 1905. 170 + viii pages, 6x9 inches; 157 diagrams; 1 plate.

The energy chart itself is shown by a plate following the title page. It consists of a system of three sets of intersecting curves: 1, isotherms, which are also lines of constant pressure and are horizontal straight lines; 2, steam-quality curves, which are only slightly curved and are approximately vertical, the one at the extreme left of the diagram representing water at the boiling point at different temperatures, and, of course, at different pressures, the one at the extreme right representing saturated steam under corresponding conditions, and the intermediate ones representing the quality of the steam when not all of the water has been converted into steam; 3, specific-volume curves, each line passing through all the points in the diagram at which the total volume of water and steam has a given value. The coördinates are not drawn, but ordinates represent temperature and abscissas represent entropy; consequently, the adiabatics or isentropics are vertical lines. This form of temperature-entropy chart offers certain advantages over the *pv* (pressure-volume) chart in the graphical solution of steam problems. At the left of the diagram, along the water line, is a temperature scale, and, outside the diagram, a scale giving the heat units in one pound of water and in one pound of saturated steam, above 32 degrees Fahrenheit; that is, the quantities represented by *h* and *H*, in the ordinary steam tables. Besides this there is another scale giving the internal energy, in B.t.u. above 32 degrees Fahrenheit, for steam; that is, the total heat in one pound of steam at the given temperature, less the heat equivalent of the external work done during evaporation.

The chemical symbol H_2O is used in the plate and throughout the book, where the water may be in the liquid or in the gaseous state, or partly in both. The words "water" and "steam," according to the ordinary usage, are mutually exclusive although by some writers water is made to include steam, or *vice versa*. The author considers that he will avoid ambiguity by using water and steam in their exclusive meanings, and H_2O as a generic term including both the others.

The author treats the subject fully, showing the application of the chart to simple and compound engine work, including designing, solving numerous problems of varied nature. He also shows that a similar chart may be constructed for any other substance besides water, providing its properties be known. The book has much merit, but leaves some things to be desired; there has been so much obscure writing on the subject of thermodynamics that perspicuity is there even more highly to be prized than in writings on other subjects. The book, although lucid in some parts, contains others which are not easy to follow for one who already has some familiarity with thermodynamics; and for one who has not, these parts are practically inscrutable. There is an apparent error on pages 26 and 27, where two methods of ascertaining the internal energy of a pound of H_2O under given conditions are described. On page 26 the solution of a numerical example given in illustration is 772 B.t.u., and on page 27, by the other method, 771.5; if a slight error in the arithmetical work be corrected, the agreement is still closer. Substantially the same figure having been obtained by two different methods, it would appear that there is a strong probability in favor of the correctness of the result; however, the internal energy of one pound of saturated steam at the given temperature, 250.5 degrees, is given in the text explaining the second method as 963.5 B.t.u., while the value shown on the scale of internal energy in the chart is 1083.5. Substituting this value, the final result is 861.6. B.t.u. The first method involves the measurement of one line in a certain geometrical construction on the entropy scale of the chart; the author makes it 0.488. The reviewer has made this construction on the chart as accurately as possible, and makes the measurement on the entropy scale 0.618, which, substituted for the other value, makes the final result 861.8, which agrees with the result attained by the other method. These results agree with the values given in the steam tables; those of the author's example do not.

The chart is very useful, even in its present form, but is drawn on too small a scale to use with facility. The author speaks of reading the dryness factor, for example, to the second decimal place, even when below 0.5, although no dryness-factor curves below 0.5 are shown on the chart. Inasmuch as the dryness-factor curves divide the isothermals into approximately equal parts, the dryness may be taken from the chart, in the aforementioned circumstances, by means of a pair of spacing dividers and a sector, or equivalent instruments, but not by simpler means; the dryness fraction of 0.47 certainly cannot be "read off the chart (Plate I)," as the author says it can, on page 33. On this and the following page, the author calls attention to the fact that adiabatic expansion does not always form water, but that, when there is already a certain fraction of water at the beginning of the expansion, the dryness may be increased. The adiabatic expansion of saturated steam always forms some water, but, on the other hand, the removal of pressure from superheated water as surely results in the formation of some steam; the dividing line is in the neighborhood of the initial dryness of 0.5. The author states that "the steam will be wetter at the end of the adiabatic expansion than at the beginning, when the dryness-fraction line through the initial point slopes from left to right, and will be drier when it slopes from right to left." This is true and would be obvious from the chart, had the author shown thereon any of the second species of curves.

In deducing the equation connecting entropy and temperature, on page 4, the author states: "The assumption will be made that the specific heat of water is constant, which it is not quite, but for practical purposes it can be considered as such." This, of course, simplifies the equation, making the entropy equal to the natural logarithm of the temperature, plus a constant; allowing for a variable specific heat, there is, of course, another term. It is probable that the author used the simplified equation in calculating the entropy values, but inspection of his "Scale of Water Heat from 32 degrees Fahrenheit (*h*)," on the chart, reveals the fact that he allowed for the variation of specific heat in this case. This is one of the instances of a lack of clearness in the text. The reader would certainly opine from the statement regarding the assumption of constant specific heat, that it is near enough for practical purposes in all cases and not simply in the calculation of entropy, which is apparently the author's meaning.

Business Items

The Wright Manufacturing Company, Detroit, Mich., are sending a very handsome desk calendar to their customers.

The International Steam Pump Company has been compelled by the steady advance in the cost of raw materials to increase the prices of its products.

F. E. Myers & Bro., Ashland, Ohio, are sending out a large wall calendar, showing the large line of goods they make, including their power pumps.

The Pittsburgh Gage & Supply Co., Pittsburg, Penn., report the recent equipment of nine power plants with their White Star continuous oiling system.

W. D. Garrett & Co., the metropolitan representatives of the Locke Regulator Company have equipped the large new engine at Columbia University, New York, with the Locke automatic engine stop and speed limit system.

The Minneapolis Steel and Machinery Company have recently opened an office at 1300 West Eleventh street, Kansas City, Mo., under the management of Louis Bendit, for the sale of their Munzel gas engine and suction gas producer.

The Haynie Machinery Company has been incorporated at Dallas, Texas, to act as manufacturers' agents. They will represent the Hooven, Owens, Rentschler Company, and the De La Vergne Machine Company. They will be glad to receive catalogs from manufacturers of power plant specialties.

Following their usual custom the Mesta Machine Company, West Homestead, Penn., were hosts at a pro-Christmas dinner ten-

dered the employees. The dinner was served during the noon hours, on Monday, December 24, in the company's extensive dining rooms tastily decorated for the occasion.

The Power Specialty Company, 111 Broadway, New York, in order to handle to better advantage the rapidly increasing demand for the Foster superheater, has within the past year opened branch offices at Boston, Chicago and Philadelphia. On January 1 it established an additional branch office in St. Louis, 1012 Chemical building.

Piqua Blower Company, of Piqua, Ohio, is being incorporated under the laws of Ohio with a capital of $50,000. This corporation will take over the interests of the Piqua Foundry and Machine Company, and will make a specialty in the manufacture of positive blowers and gas exhausters as developed by the latter company in the past two years.

The Buffalo Forge Company has recently contracted to furnish two specially designed fans for furnishing forced draft for the American stokers of the American Stoker Company, which will be used at the new steam plant of the General Electric Company, at Schenectady. The fans will be direct-connected to high-speed motors through Buffalo flexible couplings.

E. H. Kellogg & Co., 243 South street, New York, manufacturers of "Anti-corrosive" cylinder oil, have a very large sale for this brand in Great Britain. They have recently received an order from Briggs, Priestly & Sons, worsted spinners, Albion Works, Laisterdyke, England, for 50 casks and mention in the order that the quality is "to be exact with that formerly furnished."

James Bonar & Co., Inc., Pittsburg, Penn., have secured David M. Kerr, of Detroit, Mich., to represent them in that district for the sale of their Pittsburgh feed water heaters, Bonar oil filters, Bonar gage cocks, Bonar exhaust heads, etc. Mr. Kerr has made a good beginning, selling two 1500-horse-power heaters and some exhaust heads to the American Car & Foundry Co., Detroit, Mich.

In the January number reference is made to the contracts for the pumps for the Mexican Light & Power Company's stations, Mexico City, Mexico. It is stated that the Westinghouse Electric and Manufacturing Company are supplying motors for single stage Morris pumps. These pumps will be furnished by the I. P. Morris Company, of Philadelphia, Pa., who have received the contract to supply the pumps for all five stations.

The Weber Gas Engine Company, Kansas City, Mo., have recently made sales of their multiple cylinder vertical gas engines and Weber suction gas producers to the amount of 1300 horse-power. These plants operate on a number of different kinds of fuel; such as Virginia and Pennsylvania anthracite, charcoal, coke, etc. Owing to the large number of contracts on hand, the Weber Company is contemplating extensive additions to their plant at Sheffield, Mo., which will provide unusual facilities for the prompt execution of orders.

The Pittsburg Feed Water Heater Company call attention to an item in the December issue stating that a certain heater company sold an 8000 horse-power feed water heater, which was probably the largest ever made in America. They wish to amend this statement as they have sold three 12,000 horse-power heaters, two to the Carnegie Steel Company, Mingo Junction, Ohio, and one to the Youngstown Sheet and Tube Company. One of these heaters weighs 56,800 pounds, while the 8000 horse-power mentioned weighed 28,800 pounds, making a difference of 28,000 pounds.

The Peerless Rubber Manufacturing Company says that its motto and policy has been "quality maintained under any and all conditions," since the inception of the Rainbow, Peerless and Eclipse packings, and they say they will not deviate from it now. As a result they are forced on the account of high prices of crude material to re-adjust all prices again. They have served their customers and friends faithfully the past year at their own expense, and now in justice to all must make a further advance in the price of these packings. They shall be pleased to mail revised quotations on application.

The Power and Mining Machinery Company, Cudahy, Wis., in conjunction with The Snow Steam Pump Works, Buffalo, N. Y., has opened a new sales office at 719 White Building, Buffalo, where will be handled the several types of gas generating apparatus, such as the Loomis-Pettibone system, suction and pressure gas plants, built by the Power and Mining Machinery Company, and the Snow gas engines, built by The Snow Steam Pump Works. Seward Babbitt, the sales manager of the first named concern, will make his headquarters at the Buffalo office, on account of the facility for conducting business from that point.

The Buckeye Boiler Skimmer Company, Toledo, Ohio, has received a communication from the Hall & Woods Company, Lima, O., regarding the boiler skimmers installed in their plant; they say: "We draw a trifle over a gallon of heavy solid mud from the drum every day. This is very plain evidence as to what it is doing, for anyone can see from this that if this much scale is forming and the sediment is removed before it has chance to settle, the value of the skimmer is apparent. We also find that the scale that was in our boiler before the skimmer was attached is being loosened. Previous to the installation of your skimmer we were compelled to clean our boiler every week, but now once every two weeks is sufficient and our engineer thinks that after the skimmer has removed all of the old scale we will be able to run a whole month without cleaning."

The Babcock & Wilcox Company has bought from The Stirling Consolidated Boiler Company, as of December 31, 1906, its American property and interests, including all accounts and bills-receivable, and has assumed its obligations and will execute its orders and contracts for boilers and appurtenances for installation and use in the United States. The plants thus purchased will be operated in the future by The Babcock & Wilcox Company under the name and style of "The Babcock & Wilcox Company (Stirling Department)," and will be opened under the direct charge and management of the same gentlemen who have operated The Stirling Consolidated Boiler Company. The gentlemen who have been connected with The Stirling Consolidated Boiler Company in its sales department will be associated with The Babcock & Wilcox Company in similar capacities. The Babcock & Wilcox Company (Stirling Department) will manufacturing the Stirling, Aultman & Taylor and Cahall water tube boilers and appurtenances heretofore manufactured by The Stirling Consolidated Boiler Company.

Help Wanted

Advertisements under this head are inserted for 50 cents per line. About six words make a line.

WANTED—Good gas and gasolene engine salesmen. State ability, experience, etc., in first letter. Atlas Engine Works, Indianapolis.

WANTED—A first-class engineer, competent to take charge of a steam plant in large paper mill in New England. Address "Engineer," 194, POWER.

WANTED—Experienced water tube boiler erectors; permanent positions with the principal manufacturer of the United States. Address Box 193, POWER.

WANTED—High-class designing engineer for rotary engines. Salary $5000 to $10,000 per year to man who has had definitely successful experience. Box 191, POWER.

WANTED—Superintendent for factory of boiler and engine supplies, instruments, gages, and thermometers. Must be first-class man with large experience in labor-saving methods, good organizer, and must be able to show results. State previous experience and salary expected. Box 196, POWER.

Situations Wanted

Advertisements under this head are inserted for 25 cents per line. About six words make a line.

ENGINEER with 14 years' experience would like to change. Prefer chief or operating engineer. First-class references. Pacific coast preferred. Box 192, POWER.

CHIEF ENGINEER of large steam electric plant wishes to change; practical and technical education; expert with indicator and valve gears; excellent references; all letters answered. Box 178, POWER.

WANTED—By a young mechanical engineer, position as draftsman or assistant in power plant design. Technical education; salary moderate. Address E. M. Warner, 1700 Heiman St., Nashville, Tenn.

WANTED—Position as chief engineer of large gas engine installation, or assistant to designer with concern building and developing large gas engines. Have had charge of largest gas engine in the State. Satisfactory reason for change. Correspondence solicited. Box 195, POWER.

POSITION WANTED as master mechanic or chief engineer, experienced in direct and alternating current lighting and power plants up to 5000 H. P., also in machine shop and as master mechanic in manufacturing plant; American; married; age 36; at present employed, but desire to change. Box 197, POWER.

POSITION WANTED by engineer with nine years' experience in steam and electrical plants, have a thorough knowledge of both the D. C. and A. C. system of generating and distribution; extensive experience in wiring both inside and outside; understand the testing of watt meters; can perform almost any ordinary repair work on both steam and electrical machinery; am perfectly familiar with the handling of the Corliss and high-speed engines, water tube and fire tube boilers, pumps and condensers; hydraulic and electric elevators, etc.; am 29 years of age, good reference. Will go anywhere; prefer Mexico, South America, or Island possession. Address P. W. R., POWER.

Miscellaneous

Advertisements under this head are inserted for 50 cents per line. About six words make a line.

GREENFIELD VERTICAL ENGINES lead the market. Send for circular. Address Greenfield Engine Works, Harrison, N. J.

ENGINES AND BOILERS, ⅛ to 2-h.p. engine castings in sets. Model and general machine work. Sipp Electric and Machine Co., Paterson, N. J. Catalog 4c.

PATENTS—H. W. T. Jenner, patent attorney and mechanical expert, 608 F St., Washington, D. C. I make free examination and report if patent can be had, and exact cost.

PATENTS SECURED PROMPTLY—Highest references from prominent manufacturers. Write for Inventors' Hand Book. Shepherd & Parker, 512 Dietz Building, Washington, D. C.

IF INTERESTED in up-to-date water-tube boilers, investigate the merits of those manufactured by the East End Boiler Works, Detroit, Mich. See their advertisement on page 151.

DO YOU WANT machinery constructed? If so, get our estimate. Designing, pattern-making and constructing in all its branches. Address Greenfield Steam Engine Works, Harrison, N. J.

WANTED—A first-class second-hand compound condensing steam engine about 400 horse-power at 85 lb. steam pressure. Also one 200 horse-power. The Wanaque River Paper Co., 271 Broadway, New York City.

WANTED—Engineers, electricians, firemen, machinists, etc. New pamphlet containing questions asked by the different examining boards throughout the country sent free. Geo. A. Zeller Book Co., 36 So. 4 St., St. Louis, Mo. See page 137.

WE HAVE on hand a few bound volumes of POWER for the years 1902, 1903, 1904. We need the space they occupy, you need the books if you are interested in power plant engineering. In order to move them quickly we will send them to the men who send us $2 a volume; state the year you want. The purchasers pay express charges. When these volumes we have on hand are gone there will be no more. Money received too late will be returned the same day. POWER, 505 Pearl street, New York City.

For Sale

Advertisements under this head are inserted for 50 cents per line. About six words make a line.

FOR SALE—Patent No. 837,844 of December 4, 1906. Address H. O. Keferstein, 624 Fourth Ave., Brooklyn, N. Y.

FOR SALE—Controlling interest in manufacturing business in California; big future. Address Roy Howard, San Diego, Cal.

FOR SALE—A slightly used engineering library, bound ¾ leather, 10 vols.; cost $50. Will sell for $15. Address B. R. T., Box 425, Chicago.

FOR SALE—Cyclopedia of Engineering, 4 volumes, bound in half morocco, slightly used; cost $19. Will sell for $5. Address T. M., Box 425, Chicago.

WANTED—Someone to furnish money to get patents on the best boiler cleaner in the world. Will sell out. Write J. A. Geller, 1032 Water St., Kewanee, Ill.

FOR SALE—Cyclopedia of machine shop practice; four volumes bound in green morocco leather; cost $18; will take $6. Address "B. F. O.," Box 425, Chicago, Ill.

FOR SALE—A five volume set of electrical books. Bound in morocco leather. Are in fine condition. Cost new $19; will sell for $9. Address "M. R. S.," Box 425, Chicago, Ill.

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CALCULATING BUTT-STRAPPED BOILER SEAMS

Shortened Method of Determining Relative Superiority of Inner and Outer Sections

BY J. E. TERMAN

In calculating the strength of butt double-strapped joints advantage may be taken of certain short cuts to determine the weakest portion of such joints. This type of joint as usually applied to boiler construction is represented in Figs. 1, 2 and 3 for double, triple and quadruple riveting, respectively.

When it is desired to find the efficiency of a joint like the one indicated in Fig. 3, for instance, the general procedure is: First, calculate the strength of the section of shell between the rivet holes *A* and *B*; second, calculate the strength of the section of shell along *C—D* and add the strength of one rivet in single shear; third, calculate the strength of section of shell along *E—F* and add the strength of three rivets in single shear; fourth, compute the strength of all the rivets in single and double shear, respectively, included in the space *A G H B* (the rivets at the lettered points of course being considered only as half rivets). The smallest of the above results is then divided by the strength of a section of the shell of a length equal to that of the outer pitch, or the distance from *A* to *B* center to center; this gives the percentage of the strength of the joint as compared with the solid plate.

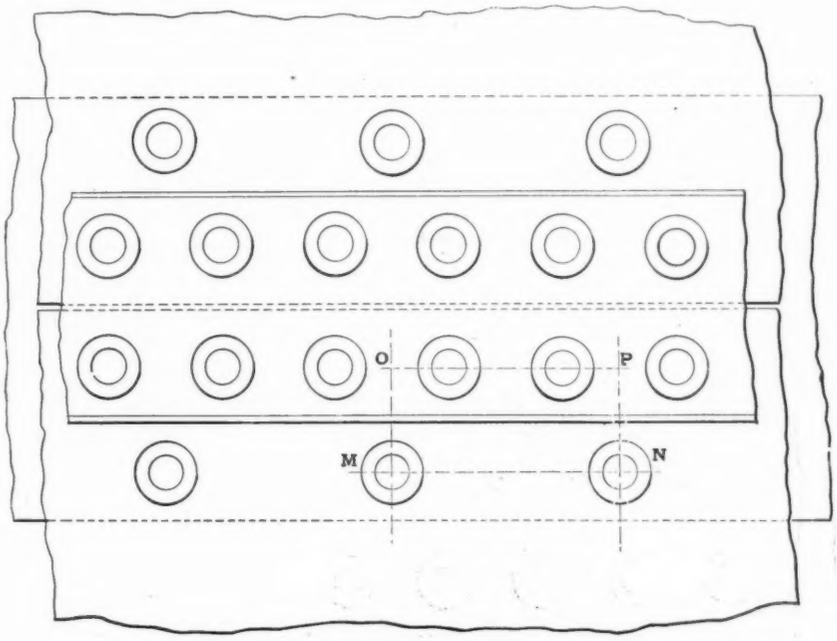


FIG. 1. DOUBLE-RIVETED JOINTS

It will be readily seen that quite an amount of labor is expended in obtaining results that are of no practical value in getting the final result, except that their relative values must be known in order to select the lowest. If this relation between the three values can be ascertained by any simpler means, it will lessen the work of determining the final result.

It will be noted that the strength of the section of shell *C—D* plus the shearing strength of one rivet would be exactly equal to the strength of section of shell between the rivet holes *A B* if the section of metal removed from the plate by a rivet hole were just equal to the shearing strength of a rivet, for this section *C—D* would then be decreased in strength below *A—B* in drilling the extra rivet hole by exactly the same amount added in shearing the rivets in the outer row. With the same relations between the thickness of plate and diameter of rivet holes, the strength of section of shell along the section *E—F* plus the strength of three rivets would also have the same value as section *A—B*, so that a joint constructed of such proportions between plate and rivet holes would be theoretically as liable to fail along any one of the three sections *A—B, C—D, E—F*.

If we assume that the plate is increased in thickness while the rivet holes remain the same, the strength removed from the inner section by drilling the extra rivet holes would not be compensated for by the shearing of the rivets in the outer rows, and the inner section would become the weaker; conversely, if the plate was decreased in thickness, the outer section would be weaker. It is obvious that if one knows whether a plate in a given joint is above or below the thickness that would make that joint equally strong in each of the sections, one can tell whether the weaker section is the outer one or the inner one.

In using this method for calculating a joint, we would first find the shearing strength of a rivet, and then multiply the thickness of the shell by its tensile strength, and by the diameter of a rivet hole, and if the strength of the rivet were less than the product of these, then the inner section would be the weaker, or otherwise the outer section would be the weaker. If several joints are to be figured, or if joints are frequently figured, a table should be prepared, as below, which gives the thickness of plate for various-sized rivet holes, at which the joint would be equally as strong at each of the three sections mentioned; the equation for getting these thicknesses would be:

$$\text{Thickness} = \frac{\text{shearing strength of one rivet}}{\text{dia. of rivet hole} \times \text{tensile strength of plate}}$$

The values assumed in this table are 60,000 pounds tensile strength for plate and 42,000 pounds shearing strength for rivets.

PROPER THICKNESSES OF PLATE TO MAKE RELATIVELY STRONG JOINTS

Diameter of Rivet Hole.	Thickness of Plate.
$\frac{11}{16}$ inch.	0.38 inch.
$\frac{13}{16}$ inch.	0.45 inch.
$\frac{15}{16}$ inch.	0.515 inch.
$1\frac{1}{16}$ inches.	0.58 inch.
$1\frac{3}{16}$ inches.	0.65 inch.

It will be evident that with the aid of such a table it is only necessary to know the thickness of plate and size of rivet hole in a given joint to determine at a glance whether the outer or inner section is the stronger. Of course, where it is necessary to compare the strength of this weaker section with the strength of all the rivets in both single and double shear,

their value must be determined in the usual way and compared with the strength of the section previously determined, but for joints constructed like Figs. 2 and 3 it will be found that the strength of the rivets generally exceeds the strength of the solid plate and unless the joint is of abnormal proportions, the strength of the rivets as a whole need not be considered; however, in the double-riveted type of joint as indicated in Fig. 1, it is well to consider this mode of failure.

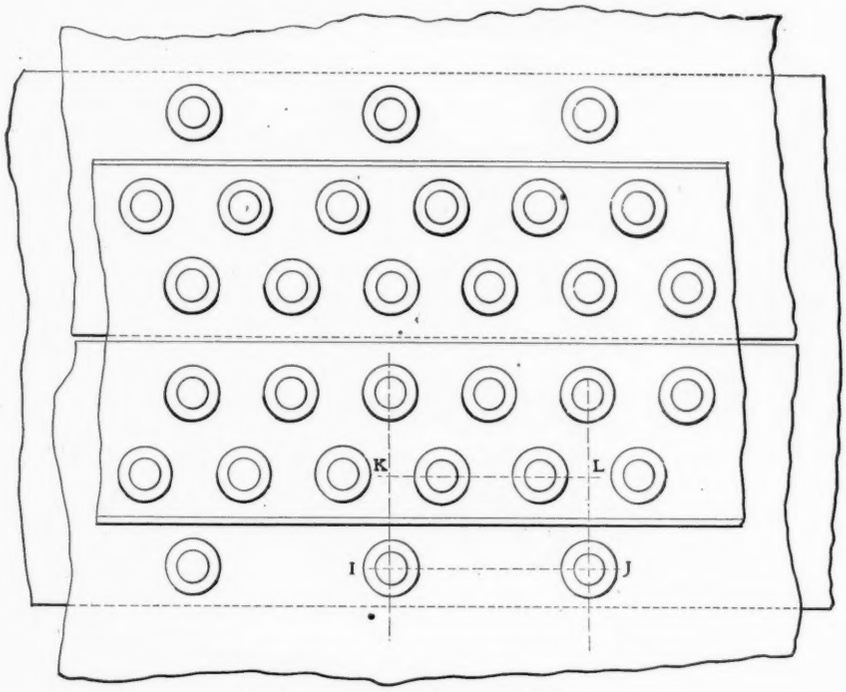


FIG. 2. TRIPLE-RIVETED JOINTS

It may be well here to point out the fact that in quadruple-riveted joints, as shown in Fig. 3, section *C—D* can never be the weakest of the three, and therefore it need never be considered in making calculations of such joints. The same methods of calculation as outlined above may be used for the double- and triple-riveted joints, and I am certain that anyone who frequently, or even occasionally has to compute the strength of such joints will never return to the laborious method of considering the strength of each mode of failure individually after trying this short cut.

There are other methods of failure of joints of this description that have not been considered, because they are not usually taken account of in the calculation of joints already constructed, as the design of such joints is supposed to render other modes of failure than those given improbable.

In the annual report of the inspector general of the steamboat inspection service it is pointed out that there are twenty-three ferryboats plying in New York waters which are more than thirty-seven years old, and that the boilers in three of them have not been rebuilt since their installation in 1863.

Large Blast-furnace Gas Engines Started Up at Pittsburg

On December 7 the first of the large gas-engine blowing units at the Edgar Thompson works of the Carnegie Steel Company was started up, and after a trying-out run of two days was put into service. We are informed that this unit has been in regular operation from the initial starting up until the present time without developing any deficiencies of construction or operation. It is a Westinghouse twin tandem double-acting engine, working on the four-stroke cycle and using blast-furnace gas for fuel, and is the first of twelve 3000-horse-power units ordered by the United States Steel Corporation. The blowing cylinders are mounted opposite to the power cylinders, and are 60 inches in diameter. The power cylinders are 38 inches in diameter and the stroke is 54 inches. The fly-wheel is located at the center of the crank-shaft, that is, midway between the two cranks. Similar engines for driving electric generators are under construction for the Steel Company; in the construction of these the generators will be mounted between the cranks, next to the fly-wheel.

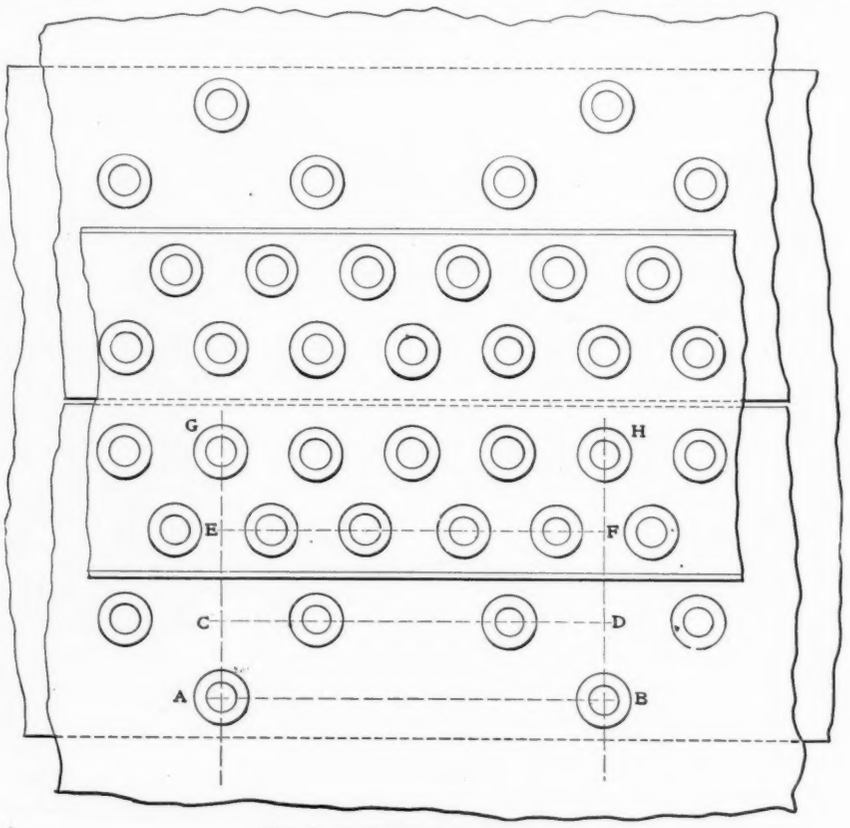


FIG. 3. QUADRUPLE-RIVETED JOINTS

Centrifugal Pumps—A Correction

On page 88 of the February number, in the final paragraph of the article on "The Centrifugal Pump," appears the statement: "One way of providing for a lower rotative speed is to build the pump in two or more stages, thereby dividing the revolutions per minute by 2, 3, 4, etc." This should have read, "thereby dividing the total head against which the pump works by 2, 3, 4, etc." The division of the head between the stages reduces the speed at which the runners must be driven, of course, but not in direct proportion to the number of stages.

COMMERCIAL ASPECTS OF OIL AS FUEL FOR POWER GENERATION

Considered in Competition with Other Combustibles Used in the Generation of Power

BY FRANZ ERICH JUNGE

VALUE, DISTRIBUTION AND QUANTITY OF PRODUCTION

In considering the relative values of the mineral and metallic products of the United States, it is found that the fuel materials aggregate about $650,000,000 annually, which is nearly double that of the output of pig iron, and about six times the value of the various precious metals produced. Of this enormous sum, which represents about 40 per cent. of the total mineral production of the country, only about one-seventh, or $95,000,000, must be credited to the output of oil, while over one-half is represented by bituminous coal, one-quarter by anthracite, and one-twentieth by natural gas. An interesting fact often lost sight of is that the oil output in the United States has a greater total value than silver and gold together.

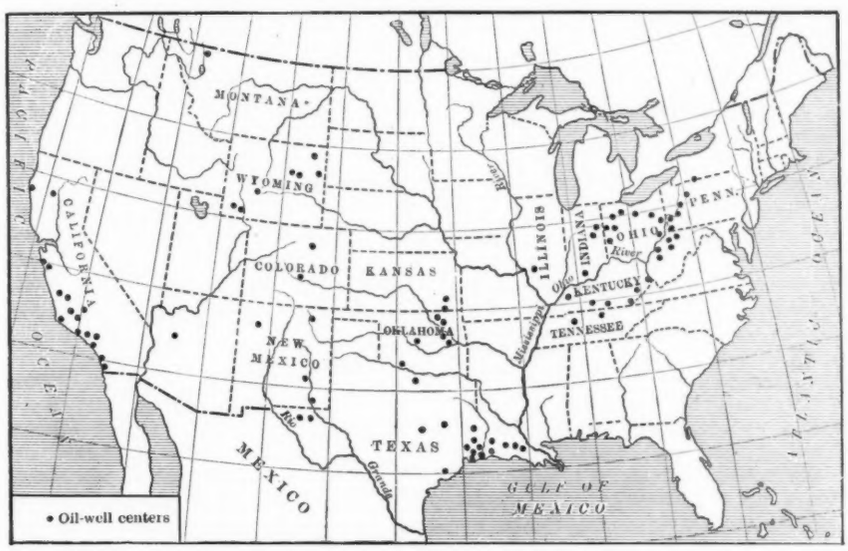


FIG. 1. DISTRIBUTION OF OIL WELLS IN THE UNITED STATES

Regarding the production of oil in this country, Fig. 1 shows that the oil centers are distributed almost equally over the whole territory, in contradistinction to the production of coal, which is chiefly limited to the eastern States. Therefore, oil as a fuel for power generation is theoretically less dependent on locality and transportation charges than coal. Owing to the fact that the production of oil was begun in the eastern States, the development of recent years shows a gradual shifting of the oil industry to the West, where virgin sources are still available. Thus California was in 1905 the main factor of production, its output being about one-fourth of the total, though for various well-understood reasons the intrinsic value of California oil is not of the highest. Then follow Ohio, Texas, West Virginia, Pennsylvania, Indiana, etc.

The heavy curve in Fig. 2 shows how the production of American petroleum increased from 3,600,000 tons, in 1880, to 13,000,000 tons, in 1903, while the dotted curve shows the fluctuations in price per ton in dollars in the course of twenty-three years. It is seen that owing to the production and sale of petroleum being controlled by the "oil trust," price and output bear a remarkably fluctuating relation to each other.

Although America is at present the greatest oil-producing country, we must, in order to arrive at just conclusions regarding the value of oil as a power factor for the industries of the world, also embrace in our consideration what is done abroad.

From 1898 to 1901 Russia marched at the head of all oil-producing countries, when, owing to the Russo-Japanese war, it dropped to second place. In the three best years the Russian production was 8,500,000, 9,790,000, and 10,920,000 tons. But in 1903 the United States output rose to 13,000,000 tons, compared to 9,700,000 tons produced in Russia during the same period. Compared with the United States and Russia, the output of all other countries in which measurements have been made are inconsiderable. Yet they are large enough to enable these (Continental) countries to emancipate themselves for a short while in case of necessity.

From 1886 to 1903 oil production in Germany increased from 10,000 to 62,000 tons, in Galicia from 43,000 to 728,000 tons, and in Roumania from 23,000 to 348,000 tons. The latest figure obtainable for the output in Germany is 110,000 tons, but it must be remembered that several other of the foreign wells are worked and controlled by German capital. Contrary to American practice the oil from Continental wells must in the majority of cases be lifted by pumps. The depth of wells varies from 130 feet to 5600 feet. Other countries where oil has been found up to this date, and enumerated according to the quantity of their respective production, are: Dutch Indian, Peru, Japan, Burma and Italy, the latter with an output of about 10,000 tons. European activities in the field of oil production are deserving more attention since a new oil trust was recently formed at Bremen, Germany, with a nominal capital of $5,000,000, for the purpose of combining all European interests to better compete with the operations of the Standard Oil Company.

GENERAL CHARACTERISTICS OF OIL

After having investigated the quantity and distribution of oil the world over, so far as it has been ascertained, it is in order to study its principal characteristics. The composition of crude oil as it emerges from the wells varies with the locality and with its geological age. But even when closely located, wells may show a marked difference in the quality of product. If the specific gravity of water is assumed as 1, then that of crude oil lies between the extreme limits of 0.78 and 0.98. The point of inflammation, or what is also called the flash point, that is, the temperature at which ignitable oil vapors are formed, lies between 24 and 35 degrees Centigrade. This must not be confused with the burning point, at which the body of the oil, or, rather, its surface, begins to ignite, such temperature being from 5 to 25 degrees above the former limit. The heat value of crude oil, that is, the number of calories contained in one kilogram, varies between 9700 and 11,700.

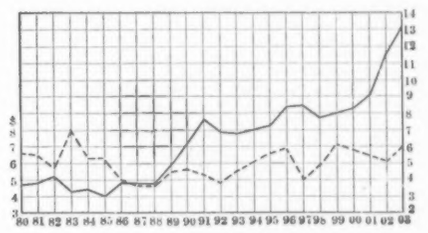


FIG. 2. OUTPUT AND PRICE OF AMERICAN PETROLEUM FROM 1880 TO 1903.

Generally speaking, crude oil forms a combination of hydrocarbons and consists principally of from 80 to 86 per cent. carbon, and from 10 to 15 per cent. hydrogen. In addition, it contains a certain number of impurities. Table 1 shows the specific gravities and boiling points of various oils from different localities. It is easily

conceivable that the utilization of oil in internal-combustion engines will be easier the lower its boiling point or the more readily it will develop oil gas or oil vapor. Therefore, the heavy oils named at the bottom of the column, and which practically form a mixture of several sorts of oil of different boiling points, are preferably treated by a process of fractional distillation, in order to separate the lighter constituents from the heavier products, or residue. By so treating crude oil we gain a series of some 15 different grades of

TABLE I. SPECIFIC GRAVITIES AND BOILING POINTS OF DIFFERENT OILS.

	Specific Gravity at 17 Deg. C.	Boiling Point, Deg. C.
Pennsylvania, 1	.8175	82
Pennsylvania, 2	.8010	74
Galicia, Sloboda	.8235	90
Baku, Bibi Eibat	.8590	91
Baku, Balachang	.8710	105
Rumania, Campina	.846	...
Alsace, Pechelbronn	.9075	135
Texas, Beaumont	.9206	150
Hannover, Oelheim	.8990	170

distillates, which can be classified in three main groups: The heavy oils, which can only be used as lubricants, or for admixture to other fuels; the characteristics of this class are: specific gravity from 0.86 to 0.96, flash point from 190 to 250 degrees Centigrade. The second class is formed by the illuminating oils, such as petroleum and kerosene, having a specific gravity of from 0.78 to 0.86, a flash point between 21 and 70 degrees, and a boiling point of over 150 degrees Centigrade. To the third class belong the unstable oils, such as gasolene, benzine, etc., which possess a specific gravity of from 0.65 to 0.75, a flash point below 21 degrees, and a boiling point below 150 degrees Centigrade.

INDUSTRIAL APPLICATION OF OIL COMPARED WITH OTHER FUELS

As was said before, the commercial utilization of these various grades of oil as fuel for generating power, regardless of whether under steam boilers or in internal-combustion engines, will be the easier so far as technical difficulties are concerned the better it can be ignited and the more efficiently it can be burnt. Thus gasolene has proven a fuel of the greatest possibilities for automobile purposes. It is obvious, however, that in addition to the above other considerations are to be made when deciding the question of applying a certain fuel to certain uses, such as price, danger of explosion in handling, respective heat value, and, before all, heat density, that is, the quantity of heat contained in the unit of volume.

The latter question is of especial importance if the fuel has to be transported in great quantities and over great distances, or if it has to be stored in tanks as a reserve. This would apply especially to the propulsion of seafaring vessels, and also to the driving of road vehicles, though to a less degree on account of the availability of supply for renewal.

Of the three classes of available fuels—gaseous, solid and liquid—combustible gases are naturally least suited for the above purposes, while fuel oils are the best. Generally speaking, the heat density of a fuel bears direct proportion to its heat value.

Another mode of comparison could be based on the respective heat value of fuels per unit of monetary value. This would again give a different ranking of the series, according to which solid fuels would occupy first place.

In Table 2 the comparison above suggested has been worked out for the majority of commercial fuels. The upper row gives the respective heating values per kilogram in calories. The middle line represents the respective heat densities, under the assumption that the value for petroleum is 1. The lower series of figures shows the number of calories that can be had for one cent, assuming prices to be the same as those that prevail on the Continent. It is seen that as far as the heat density is concerned gases are inferior to all other fuels. This is quite an important feature since the modern tendency in power distributing methods aims toward the transmission of fuels from the locality of their production, through ducts over wide distances into cities, to be used for the generation of energy. In order to increase the quantity of gas delivered per unit pipe cross-section, at the same time decreasing the dimensions and cost of pipe lines, one is obliged to compress the gas. Besides by other factors, the upper compression limit is rigidly drawn by the consideration that the negative pump work which is expended will outweigh the gain that may be expected from compression. Regarding the power required and the cost of compressing 1000 cubic feet of gas, it has been found that gas can be transmitted through a ten-inch pipe at 53 pounds pressure at the same cost as through a twenty-inch pipe at 53 pounds pressure.

In order that the same efficiency and capacity of transmission be attained with, say, illuminating gas, as when transmitting petroleum, it would have to be compressed to 1700 atmospheres. Oil gas, which is often used in districts where no coal is available for gasification, has a higher heating value than illuminating gas and a much higher heat density. Its quality and composition are largely dependent on the kind of oil gasified, and also on the process of vaporization. Every form of low-grade oil, whether crude or by-product, can be gasified in retorts heated to redness by the combustion of additional coke. The resulting gas contains from 30 to 40 per cent of the heat value of the oil and coke, and the cost of production averages ten cents per cubic meter. Its specific gravity is 0.9 kilogram per cubic meter; its density 0.6 of that of air, which is assumed at 1. Its lower heating value is about 9000 calories per cubic meter, and the amount of air required for complete combustion is 9.5 cubic meters per cubic meter. As was said, this gas is of importance in oil districts for the generation of power in gas engines, especially those of large sizes, since with direct injection of liquid fuels one cannot very well go beyond a certain capacity.

Oil gas would lend itself for transmission even better than illuminating gas on account of its greater heat density, although as compared with liquid fuel the costs of transmission are of course much higher. Thus it would have to be compressed to 950 atmospheres (compared to 1700 atmospheres for illuminating gas), in order to represent the same transmission volume as petroleum. The relation which natural gas holds to other fuels was analyzed in an article on "Some Considerations Affecting the Application of Waste Gases for Power Purposes" in the January, 1907, issue of POWER.

Now leaving aside the gasification of oil and comparing the various grades of distillates with each other, it is seen that regarding heat density or suitability for transportation, storage and transmission crude oil and petroleum occupy the first place. Among the mineral fuels hard coal is the nearest, the ratio to petroleum being 62 to 100. This is the numeric explanation for the fact that the various navies attempted to adopt oil instead of coal as fuel for raising steam in ship boilers, since with the same storage capacity the radius of action would be increased in a similar ratio. Of course, there are technical considerations besides, such as ease of handling the fuel, reduced

TABLE 2. THERMAL VALUE, HEAT DENSITY AND HEAT COST OF GASEOUS, LIQUID AND SOLID FUELS.

	Combustible Gases.				Fuel Oils.				Solid Fuels.			
	Illuminating Gas.	Oil Gas.	Producer Gas.	Blast-furnace Gas.	Petroleum.	Crude Oil.	Gasolene.	Alcohol (90 Vol. Per Cent.)	Hard Coal.	Lignite (Brown Coal).	Gas Coke.	Wood.
Heating value, cal. per kg.	9700	9900	1080	760	10,500	10,000	11,000	5600	6500	3000	7000	2800
Heat density (petroleum = 1.00)	.06	.11	.012	.011	100	99	90	56.8	62	25	37.5	11
Heat units per 1 cent.	1764	1344	.....	.....	1785	4158	1659	1306	10,794	18,270	8400	4536

personnel for attendance, etc., which promised advantages over coal. Up to this date they have apparently not been realized, however. The above refers to large vessels where comparatively great quantities must be carried. For smaller tug boats, yachts and the like, oil engines, especially of the Diesel type, have been extensively adopted in Europe. It is estimated that the range of action of a ship equipped with Diesel engines burning crude oil is two and a half times the distance run by a vessel fitted with producer-gas engines, and from four to five times the range of a steamship, for the same weight of fuel.

Considering the various fuels with respect to their monetary value, one reason why the above experiments cannot be wrought to a commercially prosperous issue for large work will be at once apparent. While one cent will buy 10,794 calories in the form of hard coal, it is only possible to get 1785 calories in the form of petroleum for the same money. This comparison also brings out by numeric analysis what is readily understood by logical consideration, namely, that of every fuel the raw product is by far the cheapest, and every effort that tends toward the direct utilization of raw fuel in engines without first submitting it to a costly cleaning, distillation or transformation process is entirely justified, so long as it is not done at the cost of reliability and efficiency of operation. Thus an engine that will burn crude unrefined oil with all its impurities as successfully as another will burn gasolene is far superior commercially, provided crude oil were available everywhere at a uniformly low price. This, however, will invariably happen as soon as there is a large demand for this fuel, and so there only remains the question of the constancy of the low price. But since under the present conditions we must reckon with the fact that the manufacture and sale of any new product which proves of importance for commercial interests will sooner or later be monopolized and controlled by some trust, whether it be alcohol or energine or what not, the question of price fluctuation is the same for every fuel, and on account of its speculative character no claims can be placed upon it for future developments. Another more definite and purely technical question is whether the existing types of engines can handle raw fuel successfully.

ECONOMIC CONSIDERATIONS AFFECTING UTILIZATION

The problem of fuel utilization and conservation assumes an entirely different aspect when considered from the stand point of the political economist, instead of as consumer of heat, light or power. Then it is obviously the most desirable practice to get maximum value of monetary returns from a certain weight of coal, oil or natural gas, since these fuels represent an irreplaceable quantity, which once exhausted cannot be restored by natural growth.

The idea of producing by-products more valuable than the original fuel and of smaller weight per unit value appears sound and is, of course, of importance to the national wealth of a people. Thus the conversion of coal into gas before combustion is a source of great income to the majority of gas industries, and in some cases one ton of coal has been made to yield products to about three times its value. The additional income thus realized depends, of course, on the markets, which may or may not favor the sale of coke, tar, sulphate of ammonia, etc., obtained through the making of gas from coal. While sulphate of ammonia is a valuable nitrogenous manure, and of increasing importance in agricultural pursuits, the value of coal tar depends greatly on the development of the chemical industries, though the possibility of satisfactorily using it as an additional fuel in gas producers and its gas oil-derivates in liquid-fuel motors will shortly increase its intrinsic value.

Similarly, the lower grades of coal, such as lignite and peat, are in modern pursuits often transformed from the raw state into more valuable by-products. The Ziegler process which is largely adopted in Germany provides for the employment of closed ovens wherein raw and wet peat is carbonized by the heat of the gases driven off, while the waste heat serves to dry the material before gasification. Through this process there are obtained from one ton of raw peat: 350 kilograms of coke, 40 kilograms of tar, 6 kilograms of alcohol, 4 kilograms of ammonia sulphate, 6 kilograms of acetate of lime, the lime and the sulphuric components being added.

Applying this line of thought to the commercial utilization of crude oil, it is obvious that by distilling the raw product and separating out lubricating oil, petroleum, gasolene, benzine, etc., the market value of one pound of crude oil can be increased to several times the value which it possesses when sold directly as a fuel for the generation of power. Owing to the high temperature (some 800 degrees Centigrade) which is necessary to completely gasify heavy oils in order that their vapors may be intimately mixed with air and ignited, the technical utilization of the lighter distillates of oil in internal-combustion engines is easier than that of the original heavy products. The volume of light and medium oils is increased through evaporation about 200 times that at ordinary atmospheric temperatures, while the minimum ratio of oil vapor to air necessary for complete combustion ranges theoretically between 1/30 to 1/25.

OIL VERSUS ALCOHOL

In conclusion, the commercial consideration is extended to a comparative estimation of the characteristics of crude oil and its refined products, on the one hand, and alcohol as its probable competitor on the other.

It has often been affirmed by statements in the technical press that alcohol has almost the same heating effect as gasolene, and that by admixing from 10 to 15 per cent. benzol, its calorific value is made even higher. Nothing can do more harm to a technical innovation than misdirected enthusiasm, and so it is better to be quite clear on this point. Fuel alcohol as used in actual practice is always mixed with some ten-volume per cent. of water, such water having an effect on the working of the engine which is theoretically harmful, but practically beneficial. By reference to Table 1 it will be seen that fuel alcohol has only about half the heating value of gasolene, while compared with crude oil the ratio is 5.6 to 10. The difference comes out as strikingly if we compare the heat density of the respective fuels which, as stated, is often the important factor in the commercial criterion, deciding the application of a certain fuel for certain purposes. While crude oil has a heat density of 99 that of alcohol is only 56.8. This means that while 0.121 liter of space is required to carry along, store or transmit 1000 calories of crude-oil fuel, 0.214, or almost the double capacity, is required for conveying the same amount of heat in form of alcohol. It is true that by the addition of benzol the calorific value and also the price of the alcohol mixture can be a little improved, but then again, leaving aside the technical difficulties, it is always preferable to depend only on one kind of fuel as the energy-transforming medium in a prime mover.

If we extend the comparison to the monetary value of the two fuels we find that in Germany 4158 calories as crude oil cost one cent, while only 1300 calories in the form of alcohol can be purchased for that price. As was said above, this ratio may improve somewhat in favor of the second claimant in this country, where greater and cheaper resources for production are available; at any rate, this will be so for some time, in order to stimulate the development of the engine and kindred industries. But who will guarantee that events will not take a similar course to that in Germany, where, after considerable capital had been invested in alcohol apparatus, the price for that fuel was suddenly raised to such an extent as to make its application, at least for large work, practically prohibitive?

It is well to bear in mind that revolutions in technical annals can only be successful when evolving from necessity and that these occasions are few and far between. More especially there is no immediate demand for the replacement of mineral oil by vegetable by-products. For, far from being exhausted, new oil wells are constantly being discovered the world over, and as far as its industrial applica-

tion is concerned, the above discussion has shown that in almost every respect oil as a fuel for internal-combustion engines is superior to alcohol. However, it is a different question whether for purposes where only small quantities of fuel are required, such as for lighting and even for power purposes in small automobile and boat motors, the superior facilities of operation and the simpler technical utilization will not conquer a wider field of usefulness for fuel alcohol than is at present possessed by oil and its distillates. It is obvious that where only a small amount of fuel is used the considerations of storage and transmission capacity, which are decisive factors in the problem with large work, do not in the smaller sizes have much weight compared to the desire for simplicity, freedom from smell, and general cleanliness. But for larger work oil is the more desirable, while for still greater capacities and a more constant load factor, the direct gasification of solid fuels in gas producers and the utilization of the gases in gas engines is the most economical method to pursue.

Just now there is being introduced in this country a new kind of fuel under the name of energine, to which reference has already been made. It is a hydrocarbon fuel obtained from crude petroleum by a special process of distillation. It is composed of about 84 per cent. carbon and 16 per cent. hydrogen, and has a calorific value of 21,630 B.t.u. per pound, compared to 18,000 in gasolene. Owing to greater heat density and thermal superiority at light loads it is possible that energine may become of some importance for use in automobile and boat motors, provided that it can be supplied at a reasonable price.

TAR OIL

Another new fuel which has found extensive adoption in Germany in competition with both oil and alcohol, and one which has largely hampered the more general application, especially of the latter, for internal-combustion engines, is manufactured and sold under the name of ergin.

Since ergin is obtained from tar by some secret process, and since it affords to the coal and gas industries a new means of disposing of the last named by-product at a high rate of profit, it will prove interesting to devote a few words to its description. Ergin is an ideal fuel for combustion engines even in the larger sizes. It is a hydrocarbon oil of almost constant composition, possessing besides a high heat value (16,500 B.t.u. per pound); also the desirable feature of being capable of suffering high compression without giving premature ignition. Another advantage consists in that alcohol motors can, without any change in piston diameter or weight of fly-wheel, be operated on the new fuel. The amount of air necessary for complete combustion is considerably higher than when using alcohol but lower than with gasolene. A peculiar feature of the combustion process is that ignition is always effected with certainty and that flame propagation is very rapid.

The consumption of ergin per horse-power-hour remains between 220 and 250 grams (about half a pound), and at a price of 1.8 cents per pound at Berlin, the cost per horse-power-hour is less than one cent, the price varying with the locality. Since ergin allows of a thermal efficiency of transformation of 30 per cent., and owing to the fact that it can be had (at present) at such extraordinarily low prices and can be supplied in practically unlimited quantities as a by-product of coal tar, it is not surprising to find that the ergin-driven engine has become a rather serious competitor not only of the small oil engine, but also of the larger suction producer-gas plants. In some of these plants it is used as a reserve fuel in case of break-down in the producer, or of coal shortage. In other plants it is used as an additional fuel for increasing the capacity of suction-gas engines, if owing to a momentary peak load a sudden increase in output is required.

Feed-water Heater Dimensions

BY N. A. CARLE

The amount of steam necessary to heat a given quantity of water through any desired range depends upon the temperature of the heating steam, the temperature of the drip or condensed water leaving the heater and the efficiency of the heat transfer between the steam and the water.

For straight-line series-installed heaters the heating steam is usually at about atmospheric pressure with a temperature of drip corresponding to this pressure and condensing about 85 per cent. of the steam passing through the heater. For heaters on the induction system the temperature of the drip is usually less and the efficiency of the heat transfer between the steam and the water is greater. An increase in the back pressure of the exhaust steam adds to its heating capacity per pound of steam.

The size of steam openings depends upon the allowable velocity of flow, which is usually taken as 6000 feet per minute for exhaust steam. The size of water openings depends upon the allowable velocity of flow for the feed-water lines, which is usually taken at 200 feet per minute.

The chart on page 141 is designed to give the size of steam and water openings for different velocities of flow and the necessary amount of steam for heating any given quantity of water through any desired range under different conditions of the heating steam.

The maximum amount of heat per pound of steam available for heating is taken as that in steam at a back pressure of 5 pounds per square inch with the drip at 100 degrees Fahrenheit, and a heat transfer efficiency of 100 per cent. The minimum amount is taken as that in steam at atmospheric pressure with a drip at 212 degrees Fahrenheit, and a heat transfer efficiency of 85 per cent. A value half way between these two is assumed to represent average conditions.

EXAMPLE:

What amount of steam under ordinary conditions will raise the temperature of 65,000 pounds of water per hour 160 degrees Fahrenheit, and what are the proper sizes of steam and water openings?

Starting with 65,000 "Pounds of Water Per Hour" read across to 160 degrees "Rise in Temperature," then up to "Average Conditions" of heating steam and across to "Pounds of Steam Per Hour," where the quantity is found to be 10,920 pounds.

For the size of feed openings start with 65,000 "Pounds of Water Per Hour," and read across to 200 "Feet Per Minute Velocity" and down to "Size of Water Opening," which is 4 inches.

For the size of steam openings start with 10,920 "Pounds of Steam Per Hour" and read across to 6000 "Feet Per Minute Velocity," and down to "Size of Steam Opening," which comes out 12 inches.

Novel Way to Prevent Engine Vibrations

According to the *Engineer*, of London, engine vibrations have been prevented from causing annoyance in the basement of a London house by the use of a novel platform. The unit is a 20-horse-power gas engine with extra heavy fly-wheel, which drives a 12½-kilowatt dynamo. A five-inch bed of concrete was first laid down, on which ten 3x4½-inch timbers were placed at equal distances apart. A series of two-inch holes were bored in the top of each timber, and a powerful spring, carrying 500 pounds with one-half-inch compression, was placed in each hole. There are forty-eight springs in all, and they support a galvanized iron tray 10 feet long, 5 feet wide and 5 feet deep. This holds the concrete base, 2 feet 8 inches thick, on which the engine rests. The exhaust and compressed-air connections are of flexible metallic tubing and the other connections of rubber tube. When the unit was started, it was found to sway longitudinally somewhat, so at each end a timber was placed.

It is announced that the plan to utilize the Victoria falls on the Zambesi river, South Africa, as a source of power for mining industries of the Rand has been revived and will be supported by a syndicate of German bankers, in conjunction with the Allgemeine Elektrizitats-Gesellschaft. The scheme includes the installation of a 600-mile overhead high-voltage line.

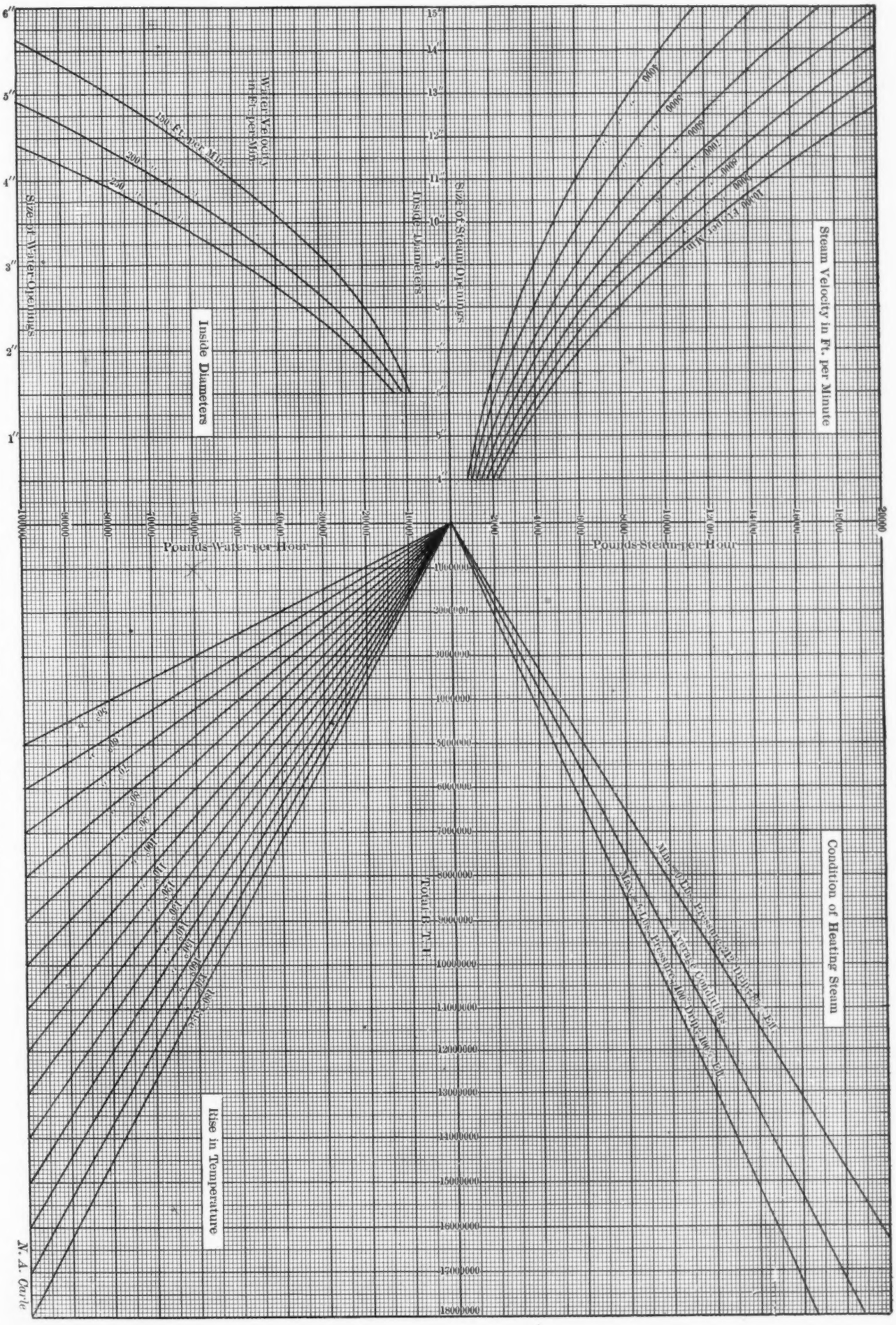


CHART FOR DETERMINING FEED-WATER-HEATER DIMENSIONS

N. A. Carle

Catechism of Electricity—XX

MEASUREMENT OF RESISTANCES

422. *Is there more than one method of measuring resistances?*

Yes; a different method must be used for measuring a low resistance from that which would be employed for measuring a high resistance if accurate results are to be obtained.

423. *What is the usual method of measuring a very low resistance?*

When the resistance to be measured does not exceed two or three ohms, the slide-wire bridge method is used.

424. *Describe the slide-wire bridge.*

The bridge itself, Fig. 91, consists of three heavy copper strips *A*, *C* and *D*, mounted on a board. Between the strips *A* and *D* is stretched a wire, usually of german silver, which is chosen on account of its high resistance. This wire is soldered firmly to the strips *A* and *D*. Below the wire is mounted a scale divided into 1000 equal parts. A slider *B* is arranged to move parallel to the wire, and has a contact piece or brush, which presses upon the wire. On the slider is fastened a pointer for indicating its position on the scale. The ends of the copper strips are provided with binding-posts for the purpose of connecting in the known and unknown resistances.

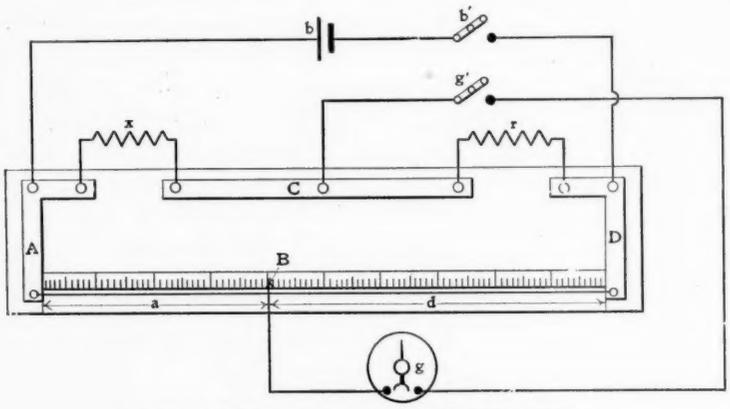


FIG. 91. CONNECTIONS OF THE SLIDE-WIRE BRIDGE

425. *How is the slide-wire bridge connected for testing purposes?*

The strips *A* and *D* are connected to a battery *b*, and a low-resistance galvanometer *g* is connected between the strip *C* and the slider. In the diagram, *b'* and *g'* represent keys, the former for opening and closing the battery circuit, and the latter for opening and closing the galvanometer circuit; *x* represents the resistance to be measured, and *r* a known resistance of the same general order as *x* (that is, if *x* is a very small resistance, *r* should be very small, etc.).

426. *Describe the method of operating the bridge.*

Upon closing the keys *b'* and *g'*, the galvanometer needle will probably be deflected; if so, this will denote an unbalanced condition in the resistances of the bridge. If the slider *B* be moved along the wire, the resistance of the wire will be added to one resistance and subtracted from the other until a balance is obtained, that is, until the points *B* and *C* are at the same potential. When this occurs, there will, of course, be no deflection in the galvanometer upon closing the keys *b'* and *g'*. The battery key should be closed a few seconds before closing the key in the galvanometer circuit, so that the current may become steady before the galvanometer is put into circuit.

Then, as the resistance of the slide wire is uniform, the value of *x*, the unknown resistance, can be obtained at once from that of the known resistance and the scale reading, by means of the formula

$$x = \frac{a}{d} \times r.$$

In this formula, *a* and *d* are the distances from the pointer on the slider to the ends of the scale, respectively, and *r* is the known resistance in ohms.

427. *Why do the symbols a and d in the formula represent lengths?*

Strictly speaking, *a* and *d* in the formula represent the resistances of the two divisions of the wire, but as the resistance of the wire is uniform the ratio of the resistances equals the ratio of the lengths.

428. *Give an example of the application of the formula.*

Suppose that with a standard 1-ohm resistance at *r* a balance was obtained when the slide was in such a position that $a = 650$ and $d = 350$. Substituting in the formula the values of *a*, *d* and *r*, $x =$

$$\frac{650}{350} \times 1 = 1.857 \text{ ohms.}$$

The value of the unknown resistance is, therefore, 1.857 ohms.

429. *Are any special precautions necessary to obtain very accurate results with the slide-wire bridge?*

Yes; in order to obtain the most accurate results it is necessary to correct, in part, for contact resistance at the ends of the german silver wire. To accomplish this the known and unknown resistance should be interchanged, and a second measurement taken. Taking the average of the two values found for *x* will tend to neutralize any error due to the above cause. The resistance coils *x* and *r* should be so placed that the current flowing through them will not affect the galvanometer inductively. This may be tested for by disconnecting the galvanometer and making and breaking the current through the rest of the apparatus, meanwhile watching for a deflection of the galvanometer.

In order to secure good contacts the wires should be scraped before a connection is made, and the binding-posts should be occasionally cleaned.

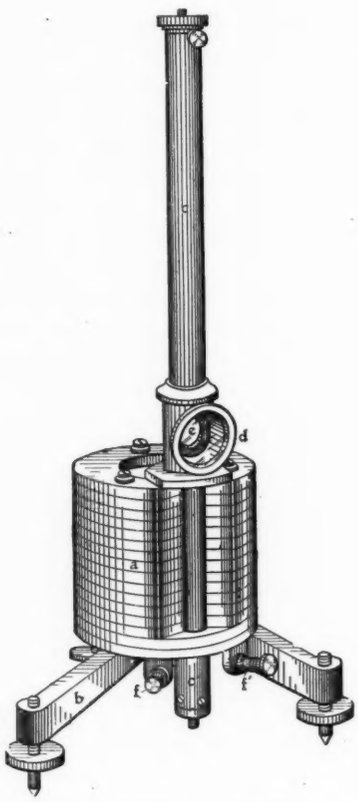


FIG. 92. D'ARSONVAL GALVANOMETER

430. *Describe the construction of the galvanometer referred to in connection with the slide-wire bridge.*

One of the common forms of galvanometers used is shown in Fig. 92. It is called a D'Arsonval reflecting galvanometer and comprises a permanent magnet *a*, which is made up of fourteen smaller magnets bolted together, one on top of the other, their like poles being adjacent. The whole is mounted on a tripod stand *b*, provided with leveling screws; *c c'* is a hollow brass cylinder, in which the movable part is suspended; *d* is an opening in the tube, covered with glass, through which the mirror *e* attached to the movable part can be seen. This tube is bolted to the magnets *a*. The binding-posts are shown at *f* and *f'*.

Fig. 93 shows the method of suspending the movable part. The coil is situated at *h*, and consists of a number of turns of phosphor-bronze wire suspended from the top of a semi-circular brass tube *k* by the phosphor-bronze wire *g*. The coil is held

at the bottom by the phosphor-bronze helix *i*. The current comes in through the spring *i*, passes through the coil to *g*, and thence out by the semi-circular tube *k*. At *j* is shown a spring which serves to clamp the coil and hold it tight, so as to prevent injury when the instrument is not in use. The mirror is shown at *e*, and the brass tube inclosing the movable part is shown at *c c'*; the latter is simply a back view of *c c'* in Fig. 92.

431. *Explain the principle upon which the D'Arsonval galvanometer operates.*

If a current be passed through a coil suspended free to swing in a magnetic field, the coil will rotate through a certain angle the magnitude of which depends on the strength of the magnetic

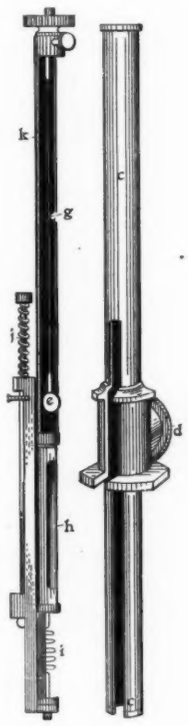


FIG. 93.

field, the number of turns in the coil, and the strength of the current passing through it. The strength of the field and the number of turns in the coil being constant, the amount of current passing through the coil is measured by the magnitude of the angle through which the coil is deflected.

432. *How is the angle of deflection ascertained?*

In order that the deflections of the galvanometer may be accurately observed, a telescope and scale are necessary. These are mounted with respect to the galvanometer mirror as shown in the diagrammatic plan, Fig. 94, where *m* is the mirror, *d* the telescope, and *s* a scale which is graduated by vertical lines about 1/16 of an inch apart. The graduations are numbered from zero up, the zero point being at the middle of the scale and the numbers increasing regularly in either side toward the ends.

When the mirror is not deflected it occupies the position shown, and the telescope, being mounted directly above the center of the scale, must be inclined slightly so that the reflected center line of the scale will be seen through the telescope in line with a fine wire stretched vertically across its center. If the mirror be deflected to the position *a* by a passage of current through the galvanometer coils, the reflected line of the scale at *u* will coincide with the center cross-wire of the telescope, while if the mirror be deflected to the position *b*, the reflected line of the scale at *v* will coincide with the center cross-wire. If the center cross-wire of the telescope coincides with the zero of the scale when no current is flowing, then when current is flowing the reflected line of the scale as seen in line with the center cross-wire of the telescope is a measure of the current producing the deflection.

433. *What precautions should be observed in mounting the galvanometer?*

The galvanometer must rest on a support unaffected by vibrations; the scale must likewise have a stable support, but it need not be as rigid as that provided

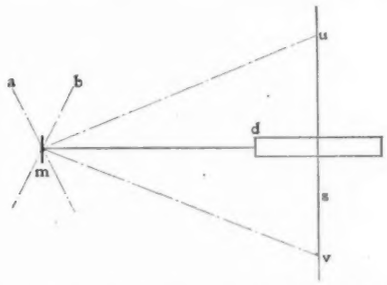


FIG. 94. DIAGRAM OF GALVANOMETER MIRROR AND SCALE

for the galvanometer. Owing to the movable coil in the D'Arsonval galvanometer being deflected only by a current passing through it and not by fluctuations in the magnetic field in which it may be placed, the galvanometer can be used in the vicinity of dynamos and motors without being affected by them. The coil, being held top and bottom, is not as sensitive to mechanical jars as in most types.

The instrument being in its desired position, the glass-covered opening should be turned squarely to the front, and leveled by using spirit levels. The object in leveling the galvanometer is not to get the instrument vertical, but to allow the needle to swing freely. The instrument should now be turned until the plane of its coil is in the magnetic meridian. A compass placed beside the galvanometer will determine this position.

434. *What is the proper way to mount the scale?*

The scale should be placed three feet from the mirror and should also be set correctly in the meridian.

The zero point of the scale, the center of the mirror and the wire in the telescope must all be in the same vertical plane. The telescope or scale should be raised or lowered until the reflection from the galvanometer mirror falls so as to be neither too high nor too low. If it be necessary to turn the mirror slightly in order that the reflected center line of the scale coincides with the center cross-wire in the telescope, this may be done by turning the suspension nut at the top of the galvanometer a trifle to the right or left. The telescope should finally be focused so that the scale and cross-wire are clear and distinct, after which the galvanometer is ready to be connected in circuit and utilized.

435. *Is any adjustment of the galvanometer necessary?*

Yes; it is necessary to vary the sensibility of the galvanometer, and for this purpose shunts are employed. Galvanometer shunts are made up in various ways, but the usual arrangement is to have three coils: one of a resistance of 1/9, one of a resistance of 1/99, and one of a resistance of 1/999 of that of the galvanometer. If 1/10 of the main current is to pass through the galvanometer, the shunt having 1/9 the resistance of the galvanometer is connected across its terminals; for 1/100 of the main current, the 1/99 shunt; and for 1/1000, the 1/999 shunt.

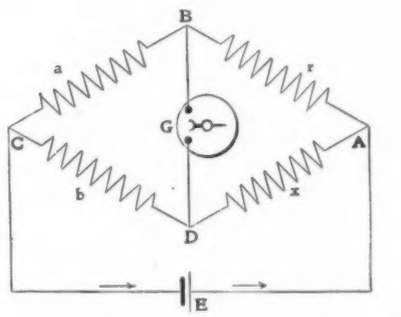


FIG. 95. DIAGRAM OF WHEATSTONE BRIDGE

436. *What is the usual method of measuring higher resistances?*

For measuring resistance above two or three ohms the Wheatstone bridge is used.

437. *Explain the principles of the Wheatstone bridge.*

The Wheatstone bridge consists of an arrangement of resistances, represented in Fig. 95 at *a, b, r* and *x*; the first three of these have known values, and the fourth one, *x*, is the resistance to be measured. A battery *E* and a galvanometer *G* are connected as shown, and by adjusting the three known resistances so that no current flows through the galvanometer, the value of the unknown resistance can be calculated by simple proportion from the known values of the other resistances.

438. *What conditions must exist in order that there be no current flowing through the galvanometer G?*

The junction points *B* and *D* must be at equal potential, so that there will be nothing to cause any current to flow across from one to the other through the galvanometer.

439. *Explain in detail what occurs when the junctions B and D are and are not at the same potential.*

Current from the battery E on arriving at the point A will divide and some will travel through the resistance a to the point B, and the rest will travel through the resistance x to the point D. If the points B and D were not connected through the galvanometer, all the current arriving at B would pass to the point C through the resistance b, and all the current arriving at the point D would find its way to the point C through the resistance r. The current, in this case, would divide in the two branches in inverse ratio to the resistances of the two paths $a - b$ and $x - r$. That is to say, if the resistance of $a + b$ was twice as large as that of $x + r$, twice as much current would flow through the path $x + r$ as in the path $a + b$.

With the galvanometer G connected as shown, another condition prevails. If the points B and D are at the same potential, no current will flow through the galvanometer. If, on the other hand, these two points are of unequal potentials, a current will flow through the galvanometer from the point of higher to the point of lower pressure, causing a deflection of the needle of the galvanometer. In order to put these two points at equal potentials, the ratio of the resistance a to the resistance b must be the same as the ratio of r to x. That is, $a \div b$ must equal $r \div x$. From this proportion is derived the equation $x = \frac{b\,r}{a}$, so that with the condition of "balance" just mentioned, and the values of a, b and r known, the value of x can be calculated easily.

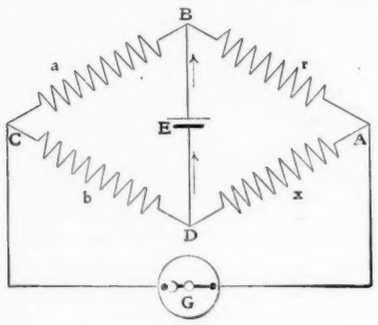


FIG. 96. WHEATSTONE BRIDGE CONNECTIONS FOR MAXIMUM SENSITIVENESS

440. *What precaution is necessary in the arrangement of the parts of the Wheatstone bridge to insure accuracy?*

Greater sensitiveness is secured by having the galvanometer connections reach from the junction of the higher two resistances to the junction of the lower two; in other words, by interchanging the positions of the battery and galvanometer so that they are connected as indicated in Fig. 96.

441. *Does this change affect the proportion between the four resistances, given in answer to Question 439 as being necessary to obtain a measurement?*

No; the proportion remains the same, and $x = b\,r$, as before.

442. *Have the resistances a, b, x and r any special names in connection with the Wheatstone bridge?*

Yes; a and b are known as the "ratio arms;" x is the "unknown resistance arm," and r is the "rheostat arm."

443. *Describe a form of Wheatstone bridge in use.*

A typical portable form of this bridge is shown in Fig. 97. Two rows of resistance coils at d and d' form the rheostat arm, and the row of coils at e constitutes the ratio arms, this row being divided into two parts for the arm a and the arm b respectively. The resistance coils cannot be seen in Fig. 97; only the brass blocks to which they are connected are visible. The box a contains besides the resistance coils the galvanometer h, and the battery.

FIG. 97. PORTABLE WHEATSTONE BRIDGE

Four terminals from the battery can be seen at f, f', f'' and f''', and connection with the battery is made at the binding-posts m and m' by means of the flexible cords k and k' and connectors l and l' attached to these cords. The binding-posts to which the unknown resistance is fastened are denoted by g and g'.

444. *Why are four battery terminals used?*

In order to obtain different battery strengths, according to the nature of the test. For measuring very high resistances more voltage is required than for measuring moderate resistances. The use of several sections of battery connected to corresponding terminals enables the operator to grade the test voltage according to the nature of the work.

445. *Show the actual circuits of the apparatus illustrated in Fig. 97.*

Fig. 98 is a plan view of the bridge with all connections indicated by dotted lines.

The rheostat arm of the bridge is divided into two rows r and r', each containing eight coils. Those in the row r have resistances of 1, 2, 2, 5, 10, 20, 20, and 50 ohms, respectively, and those in the row r' are of 100, 200, 200, 500, 1000, 2000, 2000, and 5000 ohms resistance. The resistance coils of the first group are represented at o. The numbers at the holes into which the short-circuiting plugs are inserted correspond to the resistance in ohms of the coils connected to the brass blocks. When a plug is withdrawn from a hole, a resistance equal in ohms to the number at the hole is inserted in the rheostat arm, and the total resistance in the rheostat is always equal to the sum of the numbers at the unplugged holes. The ratio arms are shown at a and b, the former being made up of three coils, of 1, 10, and 100 ohms, respectively; the other arm is composed of three resistance coils of 10, 100, and 1000 ohms, respectively. The two arms are connected together by means of four metal blocks, A, R, B and X. By inserting two plugs, one at H, and the other at H', the two arms e and e' can be connected together. By taking out these two plugs and inserting them in L and L', the relative position of the bridge arms can be reversed. Two keys shown at j and i, both in Fig. 97 and in Fig. 98, are used, the former to open and close the battery circuit and the latter to open and close the galvanometer circuit. Two of the battery terminals are shown at f' and f'', the flexible cords at k and k', and the connectors at l and l'.

446. *Describe the method of using the Wheatstone bridge.*

The resistance x to be measured is connected to the binding-posts g and g'. Plugs are inserted in H and H', and the battery connected by means of the flexible cords

k and k'. Suppose 100 ohms be unplugged in one of the ratio arms, and 10 ohms in the other; also that a certain resistance be inserted in the rheostat arms. If, now, upon pressing both the battery and galvanometer keys j and i, the galvanometer needle h is deflected toward the side marked +, there is too much resistance in the rheostat, and plugs must be inserted until, upon closing the keys j and i, the needle h remains at O. The bridge is then said to be balanced. When this condition is obtained, the sum of all the resistances unplugged in the rheostat arm multiplied by the ratio of the sum of the resistances unplugged in the A side of the ratio arm to that unplugged in the B side equals the unknown resistance x. If on first closing the keys j and i, the needle h swings toward the side marked —, it indicates that there is not enough resistance in the rheostat arm, and that more coils must be unplugged to get a balance.

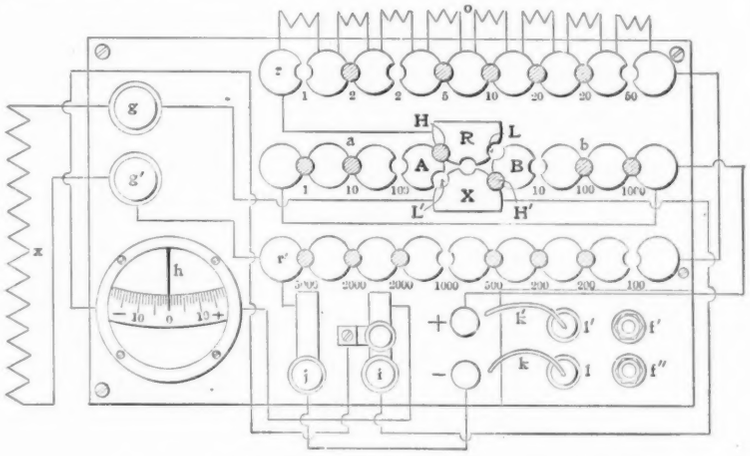


FIG. 98. PLAN VIEW OF PORTABLE WHEATSTONE BRIDGE

447. *What would be the unknown resistance x, if a balance was obtained with the Wheatstone bridge in the exact condition shown in Fig. 98?*

The shaded circles represent the holes into which plugs have been inserted, and the coils therefore cut out, while the circles shown white indicate the coils that are in circuit.

It is evident that $a = 100$, $b = 10$, while $r = 1 + 2 + 50 + 100 + 1000 = 1153$ ohms. Therefore, according to the formula

$$x = \frac{b\,r}{a}, \quad x = \frac{10}{100} \times 1153 = 115.3$$

ohms.

448. *What would have been the resistance x, if the conditions remained the same, with the exception that the plugs were taken out of H and H', and inserted in L and L'?*

The positions of the ratio arms would be reversed, so that the values would be as follows: $a = 10$, $b = 100$, and $r = 1153$. Substituting these values in the formula, the result would be 11,530 ohms.

449. *What are the limiting values of x possible to measure on the Wheatstone bridge in Fig. 98?*

The lowest value is

$$x = \frac{1 \times 1}{1000} = 0.001 \textit{ ohm}.$$

The highest value is

$$x = \frac{1000}{1} \times 11{,}110 = 11{,}100{,}000 \textit{ ohms}.$$

450. *Mention any precautions that should be observed in using the Wheatstone bridge.*

The ratio of a to b or b to a should not be greater than 10 to 1 in very accurate measurements of high resistances, as any error due to contact resistances, etc., is multiplied according to the ratio used. In any case the most reliable results are obtained when the four resistances a, b, r and x are of nearly the same value, that is, when no one of them is more than ten times the value of any other.

The brass plugs must be twisted in firmly to avoid what is called plug resistance. At the end of the test, however, the plugs should be loosened to relieve the strain on the top of the box.

The battery circuit should be closed about 10 seconds before the galvanometer circuit, so as to allow the current to become steady.

In case an exact balance cannot be obtained, the results secured for deflections of the galvanometer needle in opposite directions should be interpolated.

451. *Explain what is meant by interpolating the results obtained on a Wheatstone bridge.*

It will not generally be possible to secure exact balance, that is, absence of deflection in the galvanometer, but two values of r may be found differing by 1 ohm, which will cause the galvanometer needle to give deflections in opposite directions. The correct value of r lies between. Hence, at the smallest ratio of $\frac{b}{a}$, values of these deflections in opposite directions should be found. Thus suppose with a ratio of $\frac{b}{a}$ equal to 1/1000, 10,042 ohms in the rheostat arm give a galvanometer deflection of 3 divisions to the right of the zero mark, while 10,043 ohms in the rheostat arm give a deflection of 2 divisions to the left of the zero mark. A change of 1 ohm, therefore, causes a movement of 5 divisions. Consequently, to cause a movement of 1 division will require 1/5 ohm, and for a movement of 3 divisions 3/5 or 0.6 ohm. Ten thousand and forty-two ohms being too small by a deflection of 3 divisions, 10,042.6 ohms will, by interpolation as this process is called, be the correct resistance r to cause an exact balance.

10,000-Kilowatt Turbine-generator Units for Rapid Transit Work in Brooklyn

Electric generating units of 10,000 kilowatts capacity each have been contracted for by the Transit Development Company, of Brooklyn. This is said to be the largest single order ever placed for power-generating machinery, and the turbines are the largest thus far produced. The equipment will consist of five 10,000-kilowatt turbine-generator units, with their complement of converting, transforming and controlling apparatus, all manufactured by the Westinghouse interests.

The dimensions of the generating units are interesting. Over-all, the turbine measures 24½ feet in length, 15 feet in width, and 12¼ feet in hight above the floor level. This is equivalent to 3.8 brake horsepower (rated) per square foot occupied, or 5 2/3 brake horse-power maximum. The combined unit measures approximately 48½ feet in length, equivalent to 0.075 square feet per kilowatt rated, or 0.049 square feet per kilowatt maximum.

The turbines are designed for a steam pressure of 175 pounds at the throttle, 100 degrees superheat and 28 inches vacuum. Under these operating conditions assumed, the units are expected to sustain their full rated load continuously with a temperature rise of 35 degrees Centigrade, the power factor ranging from 90 to 100 per cent. In the event of loss of vacuum, accidental or otherwise, the turbines will automatically "go to high pressure," carrying their full rated load without the assistance of a condenser. This feature will be obtained through the use of a secondary admission valve of construction similar to the primary valve and operated by the governor in such a manner as to come into operation automatically when the overload on the machine reaches a certain point. The function of this valve is to admit steam at initial pressure to the second or further stages, and thus increase the capacity of the machine.

Speed adjustment will be practicable by a distant-control mechanism attached to the governor and operated from the switchboard. Close regulation may be obtained if desired when running alone, and when running in parallel with other machines the regulation may be changed to 3 or 4 per cent., if found desirable.

In the construction of the generator the

standard revolving-field design will be employed with frame entirely inclosed so as to facilitate forced ventilation and incidentally reduce the noise due to the high speed. Current may be delivered at either 6600 or 11,000 volts, according to the way the windings are connected up.

The condensing plant will be located, as usual, in the power-house basement directly beneath the turbines, giving a clear engine-room floor.

This new power equipment will be installed in the new Kent avenue station, Brooklyn, where three large turbine units have been in operation for some time. This station will then be entirely equipped with turbine machinery.

The Kerr Steam Turbine

By C. V. Kerr.

Turbines have been classified as impulse, reaction, velocity and pressure turbines, depending on the manner in which the energy of the steam is supposed to be used. To those who are familiar with water wheels it will be more easily understood to say that the Kerr steam turbine is built on the principle of the Pelton water wheel. It is a nozzle and bucket turbine of the purely impulse type, as compared with the guide and vane turbine of the reaction type. The most widely known of steam turbines at present are built on the principle of the water-turbine wheel. The Pelton or double-cup form of bucket gives a nearly complete reversal of the jet of steam. By compounding or dividing the turbine into several stages and the range of steam pressure into an equal number of drops, the speed of rotation is reduced

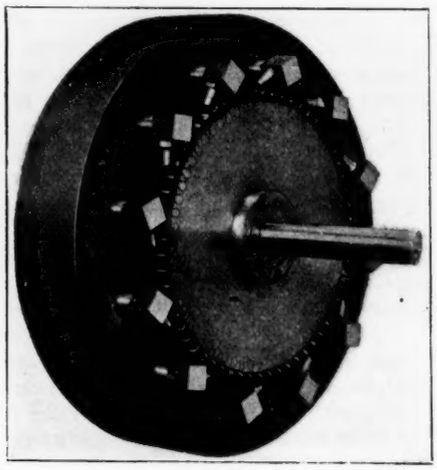


FIG. 1. TURBINE DISK MOUNTED ON SHAFT

so that gears may be avoided and direct coupling to driven machines secured. The complete reversal of the jet of steam enables the minimum number of stages to be used.

The Pelton type of water wheel has been chiefly used for high heads of water ranging from 50 feet to 200 feet. It is, perhaps, under such conditions, the most efficient of all water wheels. With high steam pressure and superheat it may be expected to show similar qualities as a steam turbine. In the old water turbine the highest efficiency was reached with low or medium heads of water for the reason that the lower the head of water and the larger the volume used the smaller the proportion of leakage through the unavoidable clearance spaces in the wheel. Hence, we find the water-turbine type of steam turbine calling for high vacuum and showing its best efficiency under such conditions. The large volume of steam in vacuum makes the leakage losses through the clearance spaces smaller in proportion. On the other hand the Pelton type of steam turbine is independent of leakage losses through clearance spaces and is consequently peculiarly adapted to the use of high steam pressure and high superheat.

The throttling type of governor is used, first, on account of its simplicity; second, the wire-drawing at light loads tends to superheat the steam and thus reduces the windage losses; third, by reducing the steam pressure within the turbine the internal resistance or windage, which is the chief source of friction losses in the steam turbine, is reduced and made nearly proportional to the load. For this and other reasons the brake horse-power developed by the turbine is found to be proportional to the steam pressure. The highest potential efficiency or the largest proportion of available heat converted into work has been found for the lighter loads. The result is a flat steam-consumption curve.

In Fig. 1 is shown one of the turbine disks mounted on the shaft in position within the circle of nozzles leading the steam from one stage to another through the castings, called on this turbine diaphragms. The shaft is made from high-grade open-hearth steel which is turned nearly to size, key-seated, drilled and fitted, then annealed under steam temperatures to remove all internal strains likely to be effected by steam heat within the turbine. Afterward the finishing cut

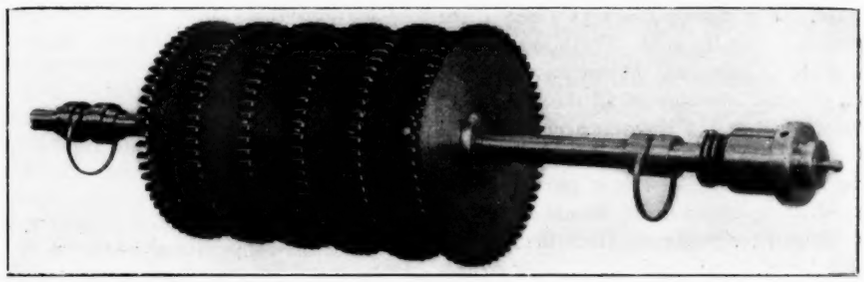


FIG. 2. SHOWING A NUMBER OF DISKS ON SHAFT

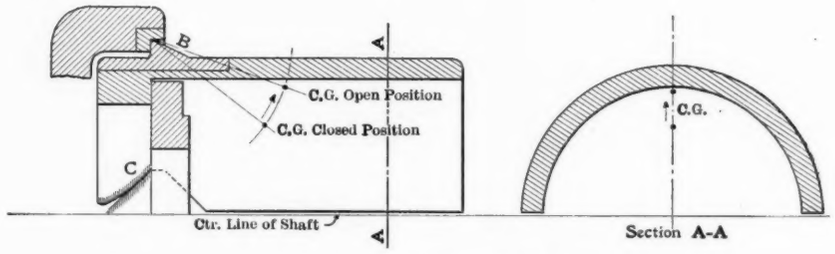


FIG. 3. ARRANGEMENT AND CONSTRUCTION OF GOVERNOR WEIGHTS

is taken on the shaft to secure true running. The disk is screwed on to the split hub against a shoulder, thus clamping the hub in place on the shaft. A lock-ring follows the disk. The double-cup buckets are drop-forged from mild steel and secured to the disk by lightly riveting in dovetail slots. After the disk is otherwise complete

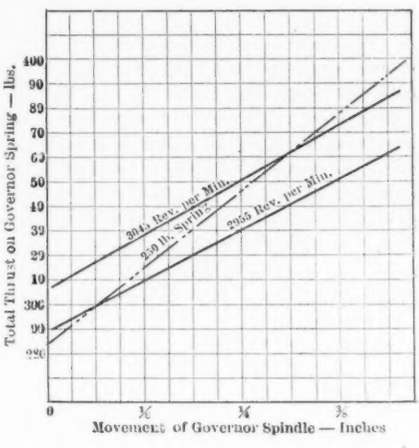


FIG. 4. TYPICAL STRESS DIAGRAM FOR GOVERNOR

it is balanced on a mandrel rolling on knife edges, which permits almost exact static balance to be secured. As the mass of metal is in a plane at right angles to the axis of rotation, the static balance obtained secures also a running balance. The slight errors unavoidably left in balancing tend to neutralize each other

when several disks are mounted at random on the same shaft.

The nozzle bodies are made from square cold-rolled steel and secured in place in the diaphragm by expanding and beading like boiler tubes. The nozzle tips are accurately formed from hexagon cold-rolled steel and screwed firmly into the nozzle body. This construction permits the possible replacing of the nozzle tips, if needed, on account of wear or of changing the size of the nozzles to secure greater or less power. The diaphragms are made from cast-iron with circular rim and arched center to withstand the steam pressure to the best advantage. Each diaphragm has a tongue on one edge and a groove on the other. When fitted together, fiber packing treated with graphite and oil is laid in the groove, thus making an elastic packing backed by a metal-to-metal outer joint. Experience has shown this joint to be easily made, easily packed and always steam tight.

The mounting of a number of disks on the shaft is shown typically in Fig. 2. At the right-hand end is shown the governor mounted on the end of the shaft and one of the ring-oiled bearing brasses. At the left is shown the other ring-oiled bearing brass, together with the locating collar on either side of the brass. There is practically no end thrust in the turbine, and theoretically none. The locating collars are used for securing and maintaining an exact alinement of the disks with the nozzles.

Fig. 3 is a diagram showing the typical arrangement and construction of the governor weights shown assembled in Fig. 2. The weight is turned from solid steel, then split, resulting in weights of uniform texture, of semi-annular form, of good running qualities with the center of gravity near to the center of the shaft. The weights are supported at three points. The hardened-steel knife edge at *B* is straight and of sufficient width for the stresses involved. At 90 per cent. on either side is a rolling contact at *C*. The curve is such that the bearing between the weight and the cam collar is always on the line of centers. Pure rolling contact is thus secured and the weight, without being fastened in position, is firmly driven by its triangular support. The outward movement of the weights compresses the spring and operates through lever connections the balanced piston valve controlling the flow of steam. The movement of the center of gravity is shown in the figure.

A typical stress diagram for the governor is shown in Fig. 4, in which the governor is designed for 3 per cent. total range of speed. In this case the centrifugal moments are reduced to the thrust on the governor spring. The result is a uniform increase in thrust with the movement of the governor spindle and steam valve. Proportions of weight may be easily found which will make these lines concave upward or convex upward, as desired. This permits the greatest stability of the governor to be secured for either light load or heavy load. In the diagram shown, equal stability at all points is given by the straight lines. The power of the governor or its ability to move the valve at any position is given by the vertical distance between the lines. Within reasonable limits this may be varied to suit the requirements. This construction secures a governor that is quick and stable, requiring no lubrication of the weights at the points of support.

In Fig. 5 are shown the working parts of the governor. The knife-edge blocks are here shown set in the rim of the cup. The spring is of the ordinary compression type with square ends seated on steel collars. The right-hand collar is movable and made long with a loose fit on the shaft to prevent binding. Motion is transmitted to the governor spindle by a transverse pin lying between the movable spring sleeve and the cam collar. This pin is free to take any circular position and rocks freely in the spindle, thus forming a universal joint.

The steam turbine is especially adapted to direct driving of high-speed machines. It is not like a patent medicine, good for anything and everything. The steam engine is a monument to the engineering profession of painstaking design and superior construction. It has qualities and applications which will secure for it, doubtless, continued use, no matter how widely useful the steam turbine may prove to be. A curious prediction was made at the time the locomotive was in-

FIG. 5. GOVERNOR PARTS FOR KERR STEAM TURBINE

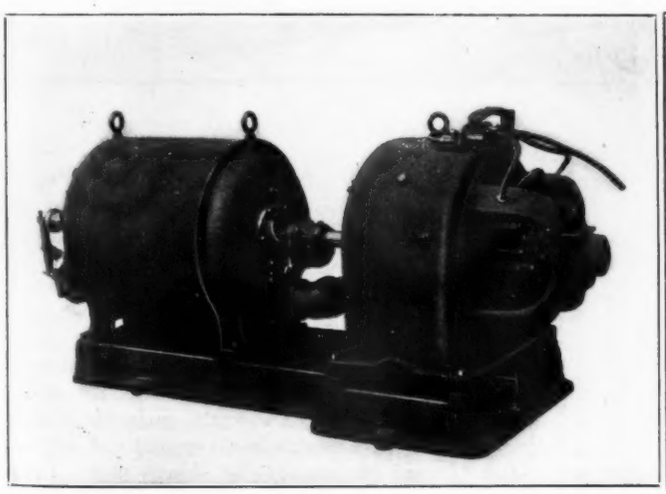


FIG. 6. KERR TURBINE COUPLED TO BURKE GENERATOR

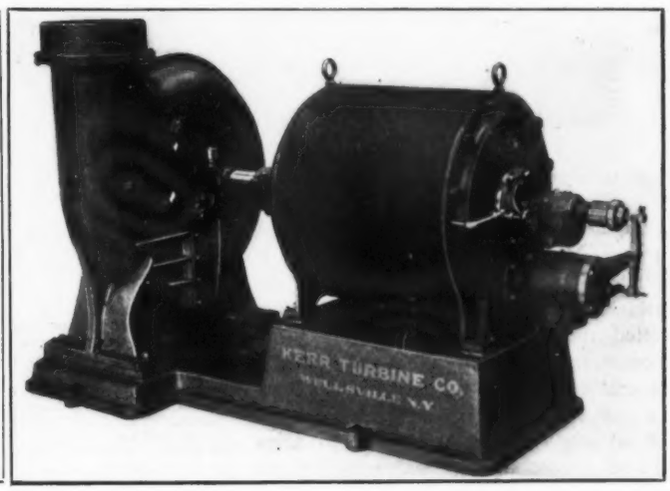


FIG. 9. KERR TURBINE AND STURTEVANT EXTRA HEAVY GAS BLOWER

troduced into England, in effect that the horse would gradually go out of use. The advent of the incandescent lamp was heralded as a supplanter of the gas jet. Gas makers now tell us their industry is in its infancy. The automobile is now making its attack on the horse and carriage. In view of history which will survive? With the steam turbine and steam engine it should be a question of "live and let live," each seeking that which it can do best.

Among the high-speed machines which the steam turbine is especially adapted to driving are generators of electric current, pressure blowers and centrifugal pumps. In Fig. 6 is shown an 18-inch six-stage Kerr steam turbine coupled to a 30-kilowatt Burke generator delivering current at 125 volts and running at 3000 revolutions per minute. The floor space is somewhat less than 3x7 feet. The unit runs with very little noise, vibration or sparking. Between the turbine and generator is shown one of the ring-oiling bracket bearings used on this type of steam turbine. The bearing case is bolted to the exhaust-end casting of the turbine. On account of the heat transmitted to the bearing by the cast-iron support, and also by the steel shaft, a mineral cylinder-oil is used in the bearing case. This permits running the bearing at any temperature likely to arise without resorting to water-cooling or other devices. The body of the turbine is made up of steam and exhaust-end castings and diaphragms which are held together by through bolts. The body of the turbine is lagged to prevent radiation and covered with a sheet-steel jacket. Leakage of steam around the shaft at the steam end of the turbine is prevented by combining metallic and fibrous packing. From the spaces between, the steam leaking through the metallic packing is piped back to a lower stage of the turbine. At the exhaust end on non-condensing work, leakage of steam is prevented by simple fibrous packing. For condensing work an inner metallic packing is also used and the space between may be filled with water to secure a high vacuum.

The lubrication of the turbine is confined to the two-ring oiling main bearings and to the governor pivot block which is provided with a reservoir of oil. There are no internal rubbing parts and lubrication is not required. The exhaust steam is, therefore, free from oil of any kind and may be used for any purpose for which clean low-pressure steam is useful. The arrangement of turbine and generator permits either machine to be removed without disturbing the other.

The result of the construction of this type of turbine and the throttling system of governing is very well shown in the following comparative tests of steam turbines given by Foster in his book on "Steam Turbines," page 162 (the last line, relating to the Kerr steam turbine, is added):

Load per cent.	100	75	50	25
	Steam efficiency per cent			
Zoelly, 350 kw	100	80.3	58.8	36.3
Rateau, 1000 kw	100	78.0	55.6	33.7
Rateau, 500 kw	100	77.2	55.0	32.0
(U.S.A.) Westinghouse, 1250 kw	100	78.5	57.3	36.0
(U.S.A.) Westinghouse, 400 kw	100	78.5	57.0	....
C. A. Parsons, 1500 kw	100	78.0	56.0	33.4
C. A. Parsons, 500 kw	100	78.0	56.0	33.7
Brown-Boveri, 3000 kw	100	77.6	56.0	33.3
Curtis, 2000 kw	100	77.2	54.2	31.4
Curtis, 600 kw	100	76.5	52.4	28.8
Curtis, 500 kw	100	76.2	51.7	28.1
De Laval, 200 kw	100	76.2	52.3	29.0
De Laval, 20 kw	100	82.0	68.0	52.0
Kerr, 50 kw	100	76.3	53.3	29.6

The above table shows results of tests made on the turbines best known at the present time. The Zoelly, Rateau, C. A. Parsons and the Brown-Boveri are turbines of European make, while the others are American. The loads are stated as percentages of the most efficient load. The nearer the steam consumption approaches the per cent. of load the better the performance. Thus at 50 per cent. load the steam economy should be as near to 50 as possible.

The 200-kilowatt De Laval turbine was governed by varying the number of active nozzles, which would not be done in ordinary service. The 20-kilowatt De Laval turbine was throttle-governed in the usual way. The 50-kilowatt Kerr turbine was throttle-governed at all loads. These are results from the first condensing turbine of this type tested which was designed as a compromise to run either condensing or non-condensing.

An application of the steam turbine which is doubtless destined to be widely useful is the driving of centrifugal pumps for handling circulating water for condensing plants, for boiler feeding and for water-works service. The speed best adapted to the turbine makes a modification of the pump design necessary for best results from the combined unit. The development of the turbine form of centrifugal pump in late years for high-pressure work makes the application of the steam turbine easy. The pump shown in Fig. 7 was designed to deliver two hundred gallons of water per minute against a head of about three hundred sixty feet. This corresponds to a pressure of something over one hundred fifty pounds, and the pump would supply at this pressure a boiler plant of about 3000 horse-power. Such units are mounted, as shown, on a cast-iron base, the pads of which are accurately surfaced. Correct alinement is thus secured and afterward maintained by dowel pins. The illustration shows an experimental unit designed for the engineering laboratory of the Armour Institute, Chicago. It is made up of an 18-inch Kerr steam turbine developing about sixty horse-power on condensing work, coupled to a Worthington three-stage turbine pump. The base is arranged so that by removing the pump and setting in place the outboard bearing pedestals, a brake-testing outfit is secured. The use of dowel pins in the setting of the pump and pedestals secures convenient handling and maintains an accurate alinement of the parts.

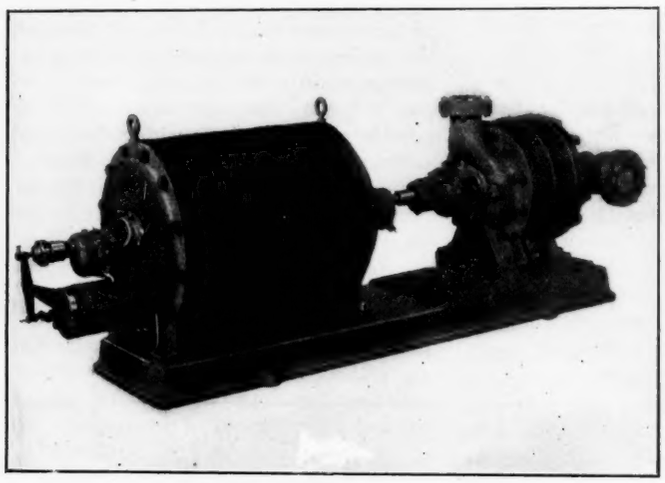


FIG. 7. TURBINE FORM OF CENTRIFUGAL PUMP

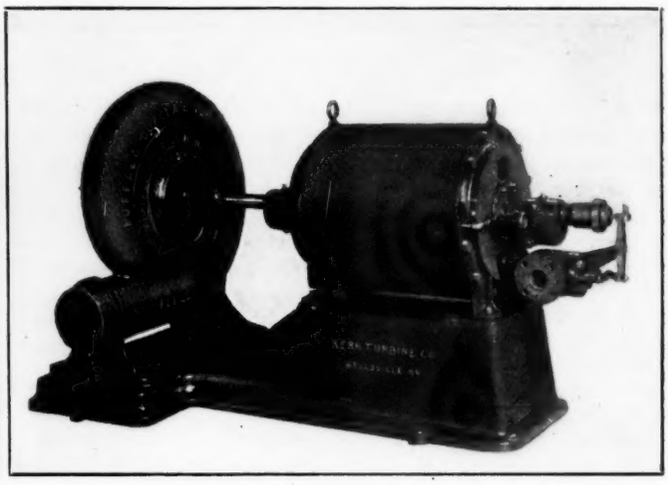


FIG. 8. KERR TURBINE AND BUFFALO BLOWER

The third application to which the steam turbine is particularly well adapted is that of driving pressure blowers for supplying blast to forges, cupolas and gas generators. In Fig. 8, is shown a blower unit made up of a 12-inch Kerr steam turbine coupled to a No. 8 Buffalo blower used for supplying blast to a 36-inch cupola. The relation between the steam pressure and blast pressure is given by the straight line in Fig. 10. The corresponding relation between blast pressure and speed is shown by the curved line which is in form nearly a parabola. In ordinary service in this case a blast pressure of seven to nine ounces, or twelve to sixteen inches water pressure, is used. The steam turbine was given sufficient power to provide for a margin over any blast pressure likely to be required.

By connecting a steam gage to the valve case below the governor valve, as shown in Fig. 11, the turbine may be run on the throttle for any blast pressure desired below the maximum or limiting pressure for which the governor is set. For instance, if twelve-inch water pressure, or about seven ounces, is desired

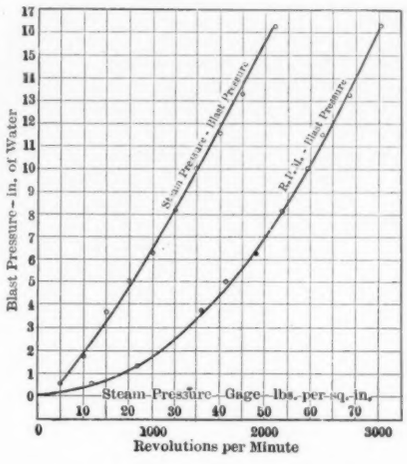

Blast Pressure – in. of Water
Steam Pressure – Blast Pressure
R.P.M. – Blast Pressure
Steam Pressure – Gage – lbs. per sq. in.
10 20 30 40 50 60 70
0 1000 2000 3000
Revolutions per Minute

FIG. 10. SHOWING RELATION BETWEEN STEAM AND BLAST PRESSURES

on the cupola, a pressure of about forty-two pounds gage would give the desired pressure at a speed of something over 1600 revolutions per minute. The high pressure of sixteen inches of water, or nine-ounce pressure, would be given by fifty-two pounds gage pressure at a speed of 2100 revolutions per minute. The cupola tender has thus the advantage of knowing the pressure of steam on his turbine, the speed in revolutions per minute and the blast pressure on his cupola, all indicated by the pressure of steam below the governor valve referred to the calibration diagram of Fig. 10. The annoyance and unreliability of the ordinary water gage is thus avoided. Here we find the advantage of the turbine over the steam engine for similar service of entirely avoiding the use of belts by this direct-coupled driving of the blower. For similar service the steam turbine will give better economy than the steam engine, together with the greater reliability obtained by avoiding the breaking and slipping of belts. The minor advantages of greater convenience in operation, of smaller space, clean exhaust and the ring-oiling bearings on both turbine and blower may be added to set forth the ideal unit for such purposes.

The water-gas process of making illuminating gas is chiefly used in this country. The process in brief consists in supplying air to a bed of fuel in a large retort or gas generator until a high temperature is obtained. Then steam is admitted which decomposes in the contact with red-hot carbon and water gas is obtained. This is afterward carbureted and becomes illuminating gas. As the decomposition of the water by the hot carbon absorbs the heat of the fuel bed the temperature is reduced in a few minutes to a point below which it is not practicable to continue the blast of steam. The air-blast is turned on and in a few minutes the fuel bed is again brought to the proper heat. This alternates at intervals of a few minutes day in and day out. For this work the turbine-driven blower, as shown by Fig. 9, has special advantages. The steam turbine is not affected by water entrained or carried over by the live steam, except by slowing down in case of an unusual amount. Under like conditions a steam engine would be in danger of knocking out a cylinder-head, or causing other damage.

The unit shown in Fig. 9 is made up of an 18-inch Kerr steam turbine, coupled to a No. 6 Sturtevant extra heavy gas blower capable of delivering air for such purposes at pressures depending on the speed, from ten to fourteen ounces or eighteen to twenty-four inches water pressure. The usual pressure is about twelve ounces or twenty-one inches of water, given by a speed of about 2800 revolutions per minute. For this gas-works service the ability of the turbine to take full steam pressure and start quickly is very useful. The unit in question has been got up to full speed from standing start by admitting full steam pressure in from five to six seconds. In emergency the turbine can be started with full steam pressure admitted to the cold turbine without danger of distress or accident. The sectional construction of the turbine and the absence of small running clearance avoid almost entirely trouble due to heat expansion.

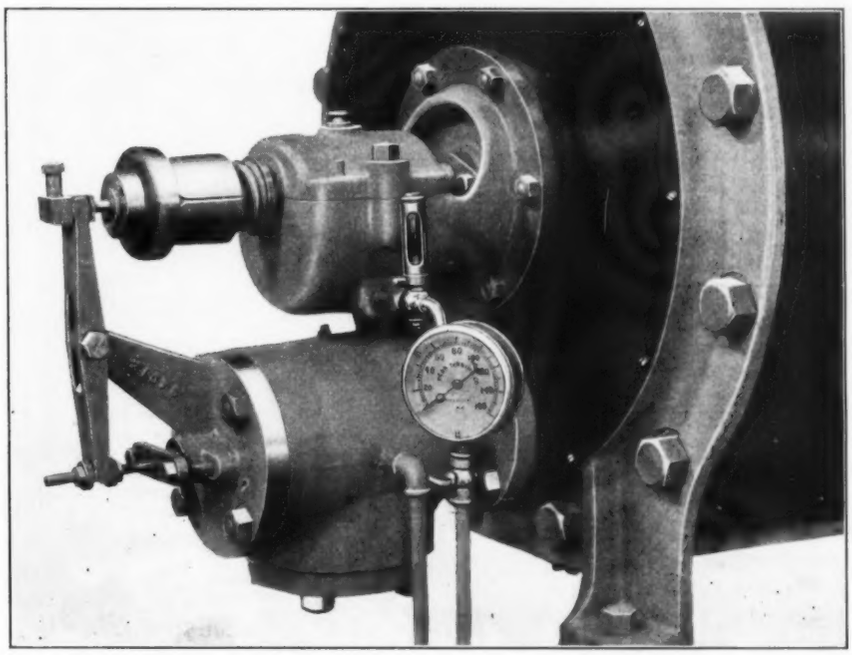


FIG. 11. GOVERNOR AND STEAM-END CONNECTIONS ON KERR TURBINE

Crude-oil Fuel

Crude-oil fuel is being used in the boiler plant of the Eagle flour mills, at Newton, Mass., at a cost comparing very favorably with that of coal. About 170 barrels of oil are burned per week at a cost of from 3.99 to 4.69 cents per barrel of flour manufactured. The fuel cost when using coal averaged 4.6 cents per barrel of flour output, not taking account of the labor cost of handling the coal and stoking. The burner used is the Hammel crude-oil burner, which uses steam for atomizing the liquid fuel, and no change was made in the furnace except to cover the grates tightly at the rear with bricks and sand, and at the front with half bricks laid loose with 1-inch air spaces; and to leave air openings on either side of the doors.—*The Boiler Maker.*

It is stated that nine-tenths of the world's supply of platinum comes from the Ural district of Russia.

THE INFLUENCE OF CENTRAL-STATION LOADS ON SMOKE PRODUCTION

Continuous Smokeless Fuel Combustion Impossible with Rapid Load Fluctuations

BY DR. CHAS. E. LUCKE

There seems to be a pretty clear conception among industrial chemists and mechanical engineers of the proper means of preventing smoke in boiler furnaces. Knowledge on this subject has been presented to the public in many papers, and in some books, as the result of scientific investigations on the one hand, and practical experimenting, more or less stimulated by municipal ordinances, on the other hand. Nearly all the work of the above mentioned kind has been directed toward the theory of combustion and the influence of certain structures of furnaces and boilers on the execution by a proper series of thermo-chemical processes to secure combustion. The result of this is that there are in the market today furnaces, with or without auxiliary apparatus, designed to carry out a certain series of processes which should lead to proper combustion without smoke, if there is no practical interference with these processes, either intentional or accidental.

Little, if any, work has been done to study the interferences which might result in smokeless combustion, even in so called "smokeless furnaces," with a view to determining whether or not these interferences, when present, are unavoidable. In some recent investigations on this subject by the writer, it appeared that a central-station load might, in some cases, constitute an interference, resulting in the production of smoke with fuel and in a furnace that would, with a different load, be smokeless. Because of the slight attention paid to this part of the subject and the small consideration given to the unavoidable nature of these interferences, by municipal ordinances, it is deemed important to present this side of the case to the profession, who may not have encountered it in their practice.

Smoke, such as issues from chimneys, consists largely of solid particles of carbon floating in the gas in sufficient quantities as to be visible, and its presence is due to two causes, the first purely mechanical and the second chemical. If the draft be extremely strong, such as is the case in high-speed locomotives, small particles of partly burned coal may be lifted from the surface of the bed and blown out of the stack, thus forming a possible smoke and constituting a considerable nuisance. This is an example of smoke produced by purely mechanical means, that is, the drawing away from the coal beds of small particles by excessive draft. When bituminous coal burns it burns in several stages. First, there is a distillation during which the volatile matter contained in the coal is separated from the coke. Next, these gases or volatile parts of the coal, which are very rich in carbon, burn above the surface of the coal bed, and finally the coke, which is left behind, burns on the grate and constitutes the main bed of the fire. If these gases are not completely burned as a result of the several causes, the carbon, which they carry in gaseous form, may separate out and become soot, which may constitute smoke; thus the second or chemical cause of smoke proper is incomplete combustion.

If the gases being distilled from bituminous coal are confined to a chamber of sufficiently high temperature, and at the same time are in contact with air also at

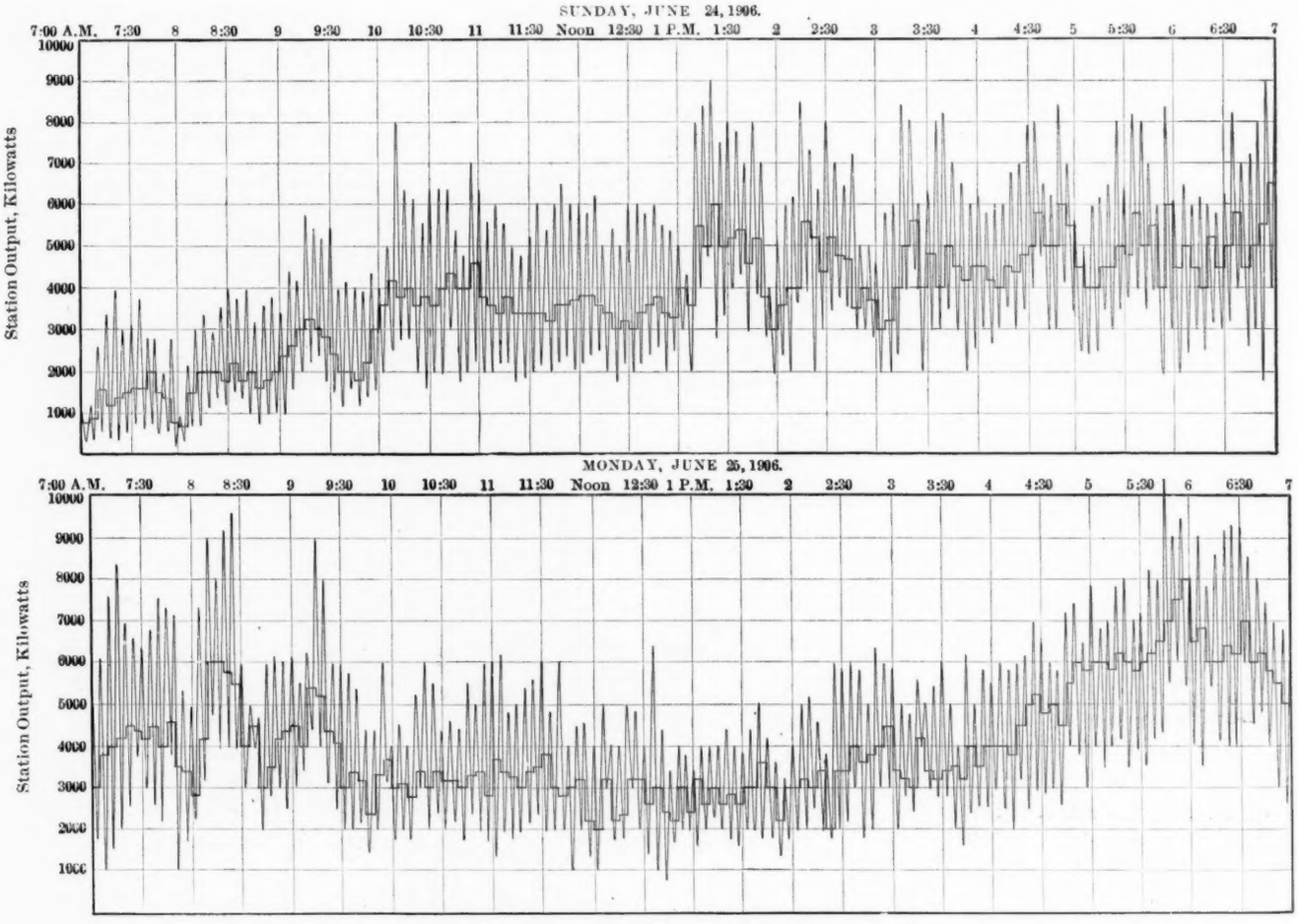


FIG. 1. TYPICAL OUTPUT CHARTS OF A RAILWAY POWER HOUSE

a properly high temperature, they will burn completely and without smoke. Combustion, therefore, to be smokeless, must be complete and demands that the entire surface, or such part of it as contains coal gas in combustion, be maintained at a high temperature and supplied with sufficient air in such a way that it

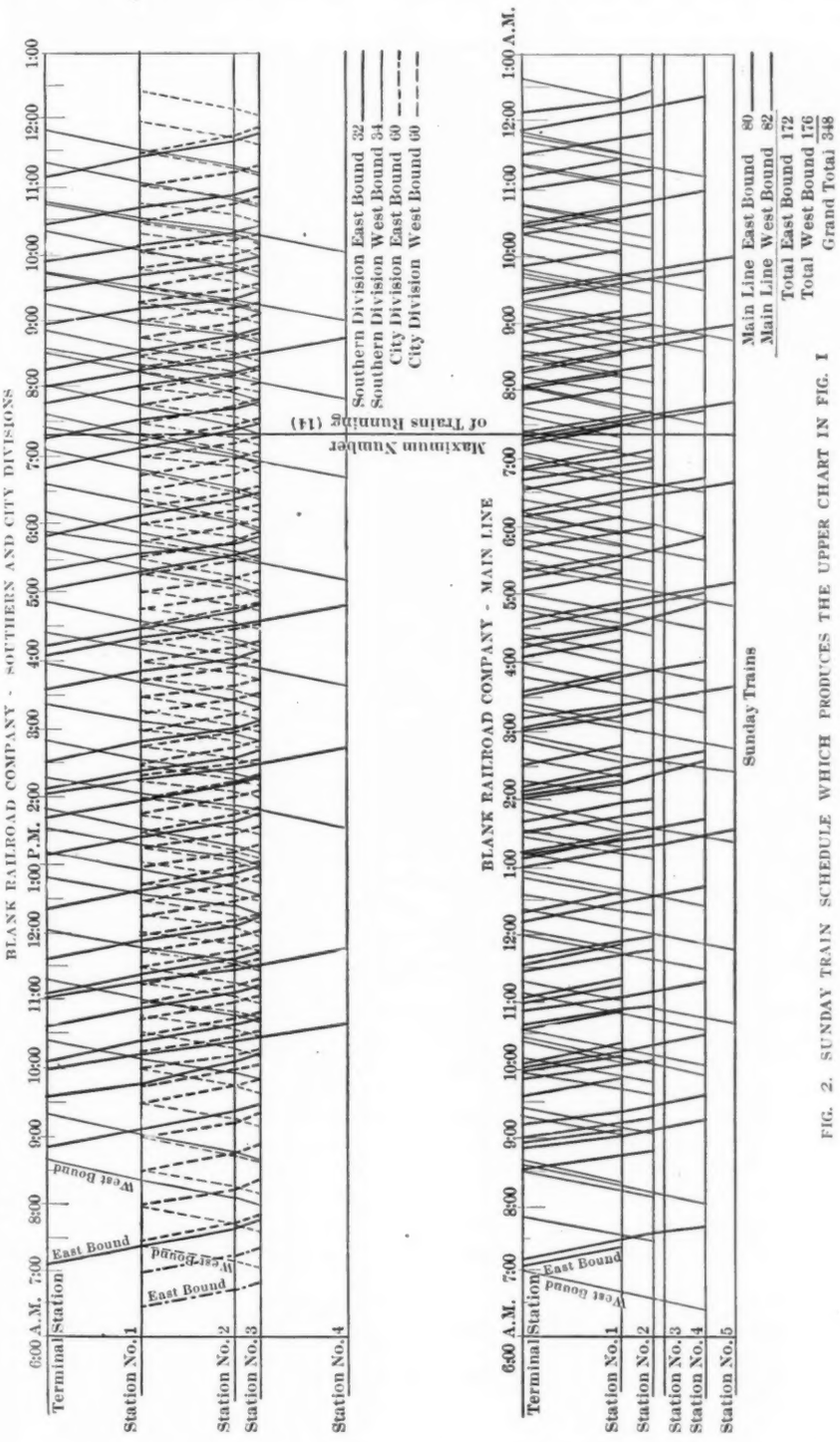

BLANK RAILROAD COMPANY - SOUTHERN AND CITY DIVISIONS
Maximum Number of Trains Running (14)
Southern Division East Bound 32
Southern Division West Bound 34
City Division East Bound 60
City Division West Bound 60
Terminal Station
Station No. 1
Station No. 2
Station No. 3
Station No. 4
6:00 A.M. 7:00 8:00 9:00 10:00 11:00 12:00 1:00 P.M. 2:00 3:00 4:00 5:00 6:00 7:00 8:00 9:00 10:00 11:00 12:00 1:00 A.M.
West Bound
East Bound
BLANK RAILROAD COMPANY - MAIN LINE
Sunday Trains
Main Line East Bound 80
Main Line West Bound 82
Total East Bound 172
Total West Bound 176
Grand Total 348
Terminal Station
Station No. 1
Station No. 2
Station No. 3
Station No. 4
Station No. 5

FIG. 2. SUNDAY TRAIN SCHEDULE WHICH PRODUCES THE UPPER CHART IN FIG. 1

may mix with the gas to be burned. Insufficient quantities of air will leave unburned particles of gas, which have passed through the flame chamber; these will be decomposed, throwing down solid carbon soot and making smoke. Should these same coal gases, during the process of combustion, be suddenly chilled either through a rush of cold air or by contact with the cold boiler pipes, they will similarly deposit carbon. It is possible to have sufficient or more than sufficient air enter the furnace, and yet if it enters in such a way as not to be uniformly mixed with the gases being distilled from the coal, in some parts of the furnace there may be large excesses of air, and in other parts great deficiency. In both cases smoke will result, on the one hand from chilling and on the other hand from lack of air.

All smoke consumers, all smokeless furnaces and smoke preventers are designed, when the inventor understands the problem, with these things in view, and no matter what the mechanism, unless the three conditions of enough air, proper mixture of air with coal gas and high temperature combustion can be carried out continuously they will fail; on the other hand, if in several types the conditions are equally well carried out, the devices are equally good. All must have a distilling chamber for the separation of the gas from the coke, a combustion chamber at a high temperature, a supply of air and means for mixing the air with the gases in such a way that every particle of gas comes in contact with the oxygen at a high temperature, and remains long enough for combustion, and means for preventing the gas flame being chilled by cold air or cool boiler plates before combustion is complete.

Incidentally, some of these smokeless furnaces or smoke-consuming devices may have longer life than others, and so be considered commercially better investments, though no better as smoke consumers. It may be also that a most excellent furnace for smoke prevention would have so short a life as to be worthless, requiring such frequent repairing as would keep many boilers out of service. It is, therefore, necessary in considering the designs of smokeless furnaces not only to be sure that they permit of perfect combustion by the laws noted, and are really smokeless, but also that their life be sufficiently long, and that the repairs be sufficiently small.

In boiler work it seldom or never happens that the rate of combustion or the rate of steam making in any one boiler is constant, because the engines need more or less steam to do their work, and the boilers must make it, and to do so must burn more or less coal. This means that the rate of combustion in a boiler furnace must be variable at will, and as the engine- and boiler-load changes sometimes occur at very short notice the furnace must permit of prompt adjustment to conditions, so that not only must the smoke-consuming furnaces be designed to carry out proper combustion and have a long life, but they must also permit of easy and rapid variation of the rate of combustion at the will of the operator to meet the demands of the steam engines supplied by the boilers.

It sometimes happens that these conditions for smokeless combustion in practical furnaces under boilers delivering variable quantities of steam become contradictory; that is to say, the demand for long life may prevent some good smokeless designs being commercially valuable, or the demand for variable rates of combustion may be impossible of fulfilment in any smokeless furnace. This is not always true, but there are such cases in existence, and there are many other cases in which smokeless combustion is possible with any kind of coal, provided the plant is properly designed.

The power house of an electrical railway system must sometimes be designed to meet conditions of service not found in any other kind of power house. These conditions may be summed up under the heading of "load characteristics." The load of a power house for an electric railway is measured by the number and weight of cars being moved and their speed at any time. The fact that the number of cars, their weight and speed measure the amount of current necessary to perform this work makes it possible also to define the load of a power station of this kind in kilowatts. As a matter of fact, the station load is measured usually in kilowatts, the engine-room load likewise is measured in kilowatts, the boiler-room load in boiler horse-power and the furnace load in pounds of coal per hour per minute or per second. Any fluctuations in the kilowatt output of the station is a direct result of the fluctuation in the number or weight of the cars being moved, and they impose corresponding load fluctuations on the steam engines, steam boilers and furnaces.

The kilowatt output load of an electric railroad station has for its chief characteristic violent fluctuations; violent because of their magnitude and suddenness. This load may suddenly vary, without any warning, from nothing to the maximum that the station can carry, and it may just as suddenly vary from maximum to nothing. The operators of the station have no knowledge whatever of what load fluctuations they will have to meet; this is under no control whatever except the motormen on the trains. As the motormen on the trains move the controller handles, the power station is drained for more power, and the men in charge of the operation of the station must meet the demand as best they can. The reason for these violent fluctuations is clear when one considers the amount of current consumed by a car or train of cars, together with the fact that many of these cars may start at the same time or stop at the same time. If there were only a single heavy train operated by a power station, the load fluctuations would probably be the greatest possible and no greater in percentage of the mean than if many heavy trains were operating from the same power station simultaneously, that is to say, all starting and stopping together with the same rate of acceleration and the same uniform speeds.

If a small number of heavy units were operated, and their starting periods should synchronize, either wholly or in part, the fluctuation of the station load would be greater than if they did not synchronize. If the number of cars or trains operated be very great indeed, and each be very small in weight, the speed slow and acceleration small, it might even happen that the load fluctuation at the station would be entirely negligible; in other words, the station might carry a constant load in spite of the starting and stopping of many cars. This is because the energy required for the starting and stopping of these many cars is such a small percentage of the mean energy required by all of the cars.

There are in operation street railway stations of all of these types, so that a street railway station load is not a definite one. It may be a fairly constant load or it may be a violently fluctuating one. It is with the most violently fluctuating load that the greatest trouble in station operation will be encountered and the greatest difficulty in securing smokeless combustion. Aside from these momentary or short-period fluctuations of a violent nature, the station must also meet fluctuations from hour to hour of the day, because the number of trains varies, there being in cities a maximum between 5:00 and 6:00 p.m., and a minimum just after midnight. Load fluctuations due to changes in the amount of current used by the trains impose corresponding load fluctuations on the boilers, and this in turn demands a corresponding variation in the number of boilers in service and in the intensity of the fire under each boiler. The changes of load from hour to hour of the day are met in the boiler-room by varying the number of boilers in use. The momentary load changes, which cannot be foreseen, can be handled only by varying the intensity of the fires under the boilers. The problem of meeting this variable rate of combustion under the boilers is one of prime importance, because the failure to regulate these fires properly means that the trains cannot be properly moved, and that the public service is unsatisfactory. In short, if it should happen to be impossible to vary rapidly the rate of combustion under the boilers in such a power station, it would likewise be impossible to handle the trains on the lines supplied by it. To serve the public properly, therefore, by carrying passengers as they want to be carried, it is absolutely necessary that such stations use furnaces in which it is possible to vary the rate of combustion as suddenly as may be necessary and through very wide limits. The possibility of rapidly controlling the fires is not only one of furnace design but of kind of coal, because only a gas-carrying bituminous coal permits of rapid changes in combustion; with anthracite this is impossible.

In railway plants, therefore, the most important thing is first to handle the load and move the cars according to schedule, because this is the basis of the charter under which the road operates. The question next in importance is smokeless combustion in the furnaces, which are subject to the interference of variable fire control. To illustrate the nature of these difficulties quantitatively some data from two large stations in everyday operation are given herewith, together with a brief record of the results.

From the load curves of station No. 1, it appeared that between the hours of 6 a.m. and 11:30 a.m., the mean load varied from about 18,000 kilowatts at 6 a.m. up to a maximum of approximately 38,500 kilowatts at about 8 o'clock. As the mean average load thus gradually rose and fell during the day, the momentary load rapidly rose and fell through very wide fluctuations every minute. Thus at 4 a.m. of this day the station was delivering 38,500 kilowatts, which dropped one minute later to 17,500 kilowatts, or less than half. Similarly at 10:49 a.m. the station was carrying 42,000 kilowatts, which five minutes later had passed through various changes and dropped to 18,000 kilowatts. The difficulties in the station can be readily appreciated when it is remembered that the engines carrying this load are capable of 7000 horse-power each, and that the full load of the station is carried on about fifty-six boilers.

In this plant, the gradual load change from hour to hour, as determined by the average traffic, is handled by putting boilers into service, or retiring them on reserve or banking the fires entirely. Thus, there are, at any time, active boilers, reserve boilers and boilers with banked fires. The engineer in charge of the fire-room and his assistants judge when reserve boilers must be brought into service and banked fires brought up to reserve, by the load-curve of the same day for the preceding week. Thus, on June 1, at 12 o'clock midnight, there were fourteen boilers in service, and forty-two boilers banked, while the output the same time was 9000 kilowatts. Comparatively similar conditions lasted throughout the night until about 4 a.m., with the exception that the output had dropped to about 4000 kilowatts. At 4 a.m. the number of boilers in service was increased to twelve by working up the banked fires. At 5 a.m. the load had not materially increased, having reached 9000 kilowatts, but the number of boilers was increased to twenty-seven in anticipation of a rapid change in output which was expected about 5:30 a.m. At 6 o'clock there were forty-nine boilers in service, and the output had jumped to 17,000 kilowatts. These forty-nine boilers were in service until 9 a.m., from which time the output had changed from 17,000 kilowatts to 43,000 kilowatts, with the same number of boilers working. Such variation in electrical output with constant boiler capacity is possible only by variations in the fires under the boilers. From 9 a.m. until 11 a.m. boilers were withdrawn from service until at the latter hour there were twenty-seven still in. This was kept constant until 2 p.m., when the number of boilers in service rose again, reaching the maximum of forty-nine at 4 p.m., at which time the output was 30,000 kilowatts. From 4 to 7 p.m. this number of boilers was again constant while the load carried varied between

30,000 and 44,000 kilowatts, met as before by the variations of the fires.

When the engines, by reason of overload, demand more steam from the boilers without warning, the only evidence the fire-room force of men have of the fact is the falling steam pressure, and this can be remedied only by forcing the fires. Should the load suddenly decrease, the effect at the boilers is a sudden rise in the pressure, and possibly a blowing off of the safety-valves. This high pressure is brought down again by closing the air-supply doors of the furnaces, by opening the side doors and closing the hand damper to chill the boiler. All emergency changes of load, that is to say, all changes that come without warning, are met in the fire-room by the stoker assistant and directed by the water tender. But little direction is given in these emergency cases on account of the shortness of the time, and it is not needed because the men must be and are trained to the work.

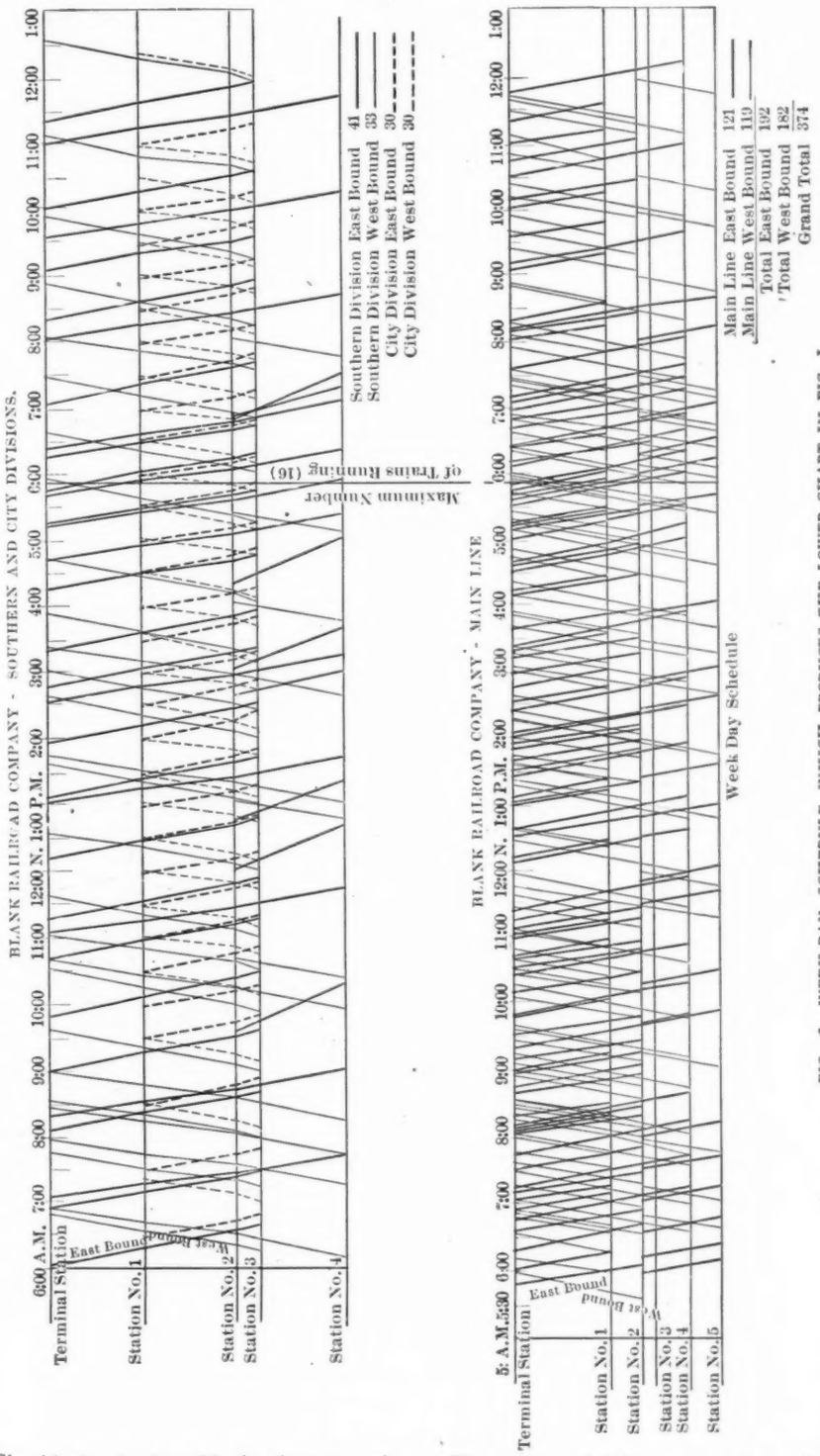

BLANK RAILROAD COMPANY - SOUTHERN AND CITY DIVISIONS.
Maximum Number of Trains Running (16)
Southern Division East Bound 41
Southern Division West Bound 33
City Division East Bound 30
City Division West Bound 30
6:00 A.M. 7:00 8:00 9:00 10:00 11:00 12:00 N. 1:00 P.M. 2:00 3:00 4:00 5:00 6:00 7:00 8:00 9:00 10:00 11:00 1:00
Terminal Station
East Bound
West Bound
Station No. 1
Station No. 2
Station No. 3
Station No. 4
BLANK RAILROAD COMPANY - MAIN LINE
Week Day Schedule
Main Line East Bound 121
Main Line West Bound 119
Total East Bound 192
Total West Bound 182
Grand Total 374
5:30 A.M. 6:00 7:00 8:00 9:00 10:00 11:00 12:00 N. 1:00 P.M. 2:00 3:00 4:00 5:00 6:00 7:00 8:00 9:00 10:00 11:00 1:00
Terminal Station
East Bound
West Bound
Station No. 1
Station No. 2
Station No. 3
Station No. 4
Station No. 5

FIG. 3. WEEK-DAY SCHEDULE WHICH PRODUCES THE LOWER CHART IN FIG. 1

The stacks of this power station show smoke at times and at other times do not. If any smoke does exist it may be attributed either to improper coal or bad design of furnace, or to the interference with the proper operation of the furnace. The facts that the load requires some bituminous coal because of its gas-carrying capacity, and that the furnace is capable of burning this completely without smoke at times, seems to prove beyond question that the trouble is not so much with the furnace as with interference with its proper operation. It might be said that the furnace should be redesigned to meet these peculiar load conditions of variable combustion, and I believe this is true, but it has so far proved to be impossible. When the precise actions in a furnace, otherwise well designed to enable it to carry this fluctuating load, are as described, it seems likely that it will continue to be impossible for at least a long time to come.

The furnaces used in this power station are of a commonly-accepted type: Roney stokers set under Babcock & Wilcox boilers. The rate of combustion in these furnaces is controlled in several ways by the stokers. At the entrance to each stack is placed a damper under the control of the steam pressure. When the steam pressure rises too high this damper is automatically closed, and the rate of combustion correspondingly decreases. In order that no one boiler may be kept out of service each is fitted with a separate hand-controlled damper. Besides these two damper controls, the automatic and the hand, the stoker operator may retard combustion by shutting off air at the front of the stoker grates or by admitting cold air through the side doors at the base of the stoker. He may accelerate combustion by reversing the above processes as well as by pushing coal down on the coke bed by a bar poker. Besides the construction mentioned these boilers are also fitted with eight steam jets for each stoker, of ⅛-inch diameter.

The coal fed to the stokers is crushed to a uniform size before it is stored in the bunkers at the top of the house. From the bunkers it passes automatically to the stokers through large pipes, reaching the stokers in small and fairly uniform pieces. These large pipes discharge into a hopper on top and in front of the stoker. From this hopper the coal is fed down the inclined grate bars by a pusher plate in a perfectly uniform manner, while the feed at any time is uniform, as mentioned. It is capable of considerable variation by the stoker operator, who adjusts it to the rate of combustion. If the rate of feed be too fast, unburned coal will be discharged under the base of the stoker into the ash-pit. If the feed be too slow for the rate of combustion ashes will form part way down the grate, making a certain part of the grate ineffective.

The coal immediately after being pushed into the fire chamber from the hopper lodges under a brick arch, where the temperature is high enough to distil the gas from the coal, but not high enough to burn it. At the front of the brick arch above the coal is a row of holes supplying air, which mingles with the gas leaving the coal. As this mixture

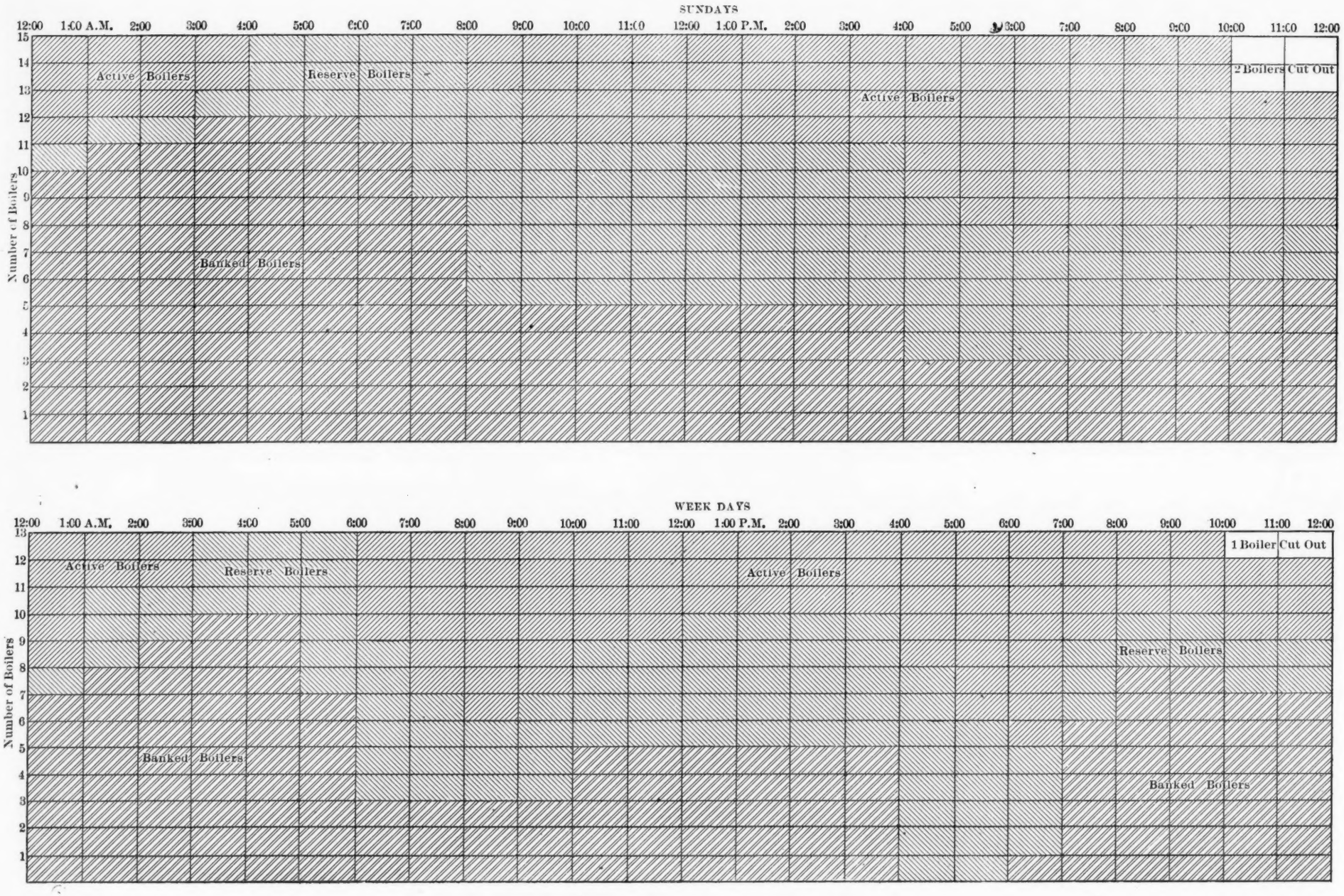

SUNDAYS
12:00 1:00 A.M. 2:00 3:00 4:00 5:00 6:00 7:00 8:00 9:00 10:00 11:00 12:00 1:00 P.M. 2:00 3:00 4:00 5:00 6:00 7:00 8:00 9:00 10:00 11:00 12:00
Number of Boilers
15 14 13 12 11 10 9 8 7 6 5 4 3 2 1
Active Boilers
Reserve Boilers
Active Boilers
2 Boilers Cut Out
Banked Boilers
WEEK DAYS
12:00 1:00 A.M. 2:00 3:00 4:00 5:00 6:00 7:00 8:00 9:00 10:00 11:00 12:00 1:00 P.M. 2:00 3:00 4:00 5:00 6:00 7:00 8:00 9:00 10:00 11:00 12:00
Number of Boilers
13 12 11 10 9 8 7 6 5 4 3 2 1
Active Boilers
Reserve Boilers
Active Boilers
1 Boiler Cut Out
Reserve Boilers
Banked Boilers
Banked Boilers

FIG. 4. BOILER-SERVICE RECORD CORRESPONDING TO OUTPUT CHARTS IN FIG. I

of gas and air passes forward under the arch it becomes hotter and hotter, while at the same time it is agitated by the eight steam jets, before mentioned. There is, therefore, a mixture of coal gas and air, further mixed by the steam, passing forward into a gradually hotter chamber under the brick arch, which prevents any chilling of the mixtures and the flame by the cool boiler tubes. These conditions are extremely good for smokeless and perfect combustion. Beyond the brick arch the bed of the grate is covered by nothing but coke, from which smoke making is practically impossible under light drafts. Thus, in this stoker there are carried out the three important principles for perfect combustion, and this fact has given the stoker in question the reputation it has.

If it is required to feed the coal too fast by reason of the demands of the engines, green coal may pass beyond the coking arch and so cause smoke through the chilling of the unburned gases against the tubes. Should cold air enter the furnace chamber at any time through the ash-pit at the bottom of the stoker, this cold air may so chill the burning gas as to make smoke. Should the demands of the load ever make it necessary to push green coal down on the coke bed to keep up the steam pressure, smoke will result also. Should the feeding of the coal be a little uneven and holes burn through the fires as a result, these must be covered with green coal and smoke will again result. Similarly, if the coal should burn faster next to the brick walls of the furnace, as it actually does, or should it be fed slower along the side, as is also the case, then there may appear at the sides holes which must be covered by green coal by the stoker operator. In all of these cases smoke will result. Again, when the boilers are carrying a uniformly good load, should this suddenly decrease it may be necessary to close the damper and open the air-chilling doors; then, as the coking will continue under the coking arch because the chamber is hot enough for this, the gases being distilled will not burn because of insufficient air and too low a temperature. This will again cause smoke. The sudden and large load changes of this plant impose many of the above smoke-making conditions on the fire, and smoke is an unavoidable consequence.

It appears, therefore, that although in this stoker conditions for smoke prevention are good for uniform loads and proper rates of feed, there are conditions continually arising with rapidly varying loads that require such handling of the stoker that smoke must result. Smoke, therefore, will result from this stoker and from any stoker when the best conditions for combustion are interfered with, as explained. Some of the conditions for smoke, but the minor ones, are due to a slight imperfection in the stoker, which like all mechanical apparatus is always being improved.

Another large power station, No. 2, though not so large as the former, operating somewhat heavier trains and fewer in number, has the load curve given in Fig. 1 for certain days of the year. The vertical scale represents kilowatts and the horizontal one the time of the day. The stepped curves in this chart represent what is ordinarily taken as the five-minute readings for estimating the kilowatt-hours output for the day. The zigzag line is the load curve for two and one-half minute readings. It is very plain from these curves that five-minute readings give no very clear idea of momentary fluctuations, as is shown by the differences between the five-minute readings. This load curve results from a train schedule illustrated in Figs. 2 and 3. The variation of mean load is handled in this station, as in all others, by taking off boilers and putting them on and the boilers are in three classes, active, reserve and banked. The chart, Fig. 4, shows the boiler work of the entire plant for an average week day and Sunday. Momentary fluctuations must be handled by the fires and they are handled precisely as in the other station described.

If a fluctuation be of extremely short duration, there may be enough energy in the steam between the engine and the boilers and in the hot water in the boilers to carry the fluctuation, provided an appreciable pressure drop in the main, necessary to release this energy, will not interfere with the operation of the plant. If for any reason, such as the characteristics of alternating-current machines operated from the station, the rate of speed must not change more than so much, then, as a consequence, the allowable pressure drop during an overload is likewise limited; in the station under consideration this is 10 pounds. If, therefore, a momentary overload in this station be of such short duration and moderate intensity that it can be met by the energy liberated in hot water and reserve steam by a 10-pound pressure drop, then the fires will need no manipulation whatsoever. If the fluctuations be of longer periods than this or of greater violence, such as would cause a greater pressure drop than 10 pounds in very short spaces of time, then the fires absolutely must be manipulated and excess coal burned, with some resulting smoke. Long-period fluctuations, such as the variation of the mean load curve through the day, are met, as explained, by putting boilers in and out of service. These three kinds of load fluctuations are possible with a corresponding provision in the means for their handling, and one of them requires manipulation of the fires so as to interfere with smokeless conditions. This is a violent fluctuation of short period, but longer than can be met by the reserve energy in the steam and water with the allowable steam pressure drop.

If by any device, method or system, the load fluctuations on the engines could be prevented from imposing similar fluctuations on the furnaces, then all of the difficulties producing smoke, as described above, would disappear. The problem is very much indeed like the problem of regulation. In this case the actual resistance of the engine is continuously varied, and the problem of regulation resolves itself into a problem of instantaneously adjusting the driving force so that it may always equal the resistance. We have never succeeded in doing this, and never will. But by the introduction of fly-wheels, in connection with other devices, we have succeeded in adjusting the mean effort for short periods of time, such as one minute, to the mean resistance for the same period of time.

If some sort of thermal-energy fly-wheel or electrical-energy fly-wheel could be applied to the central-station load, so that it would alternately absorb and give out proper amounts of energy, the demand on the boilers might be made less variable and the rate of combustion in the furnace correspondingly less variable. For this purpose storage batteries have been used in direct-current plants of small size, to take up fluctuations of electrical load and impose a less variable load on the engines. In some situations, however, storage batteries are out of the question. Similarly, large tanks of superheated water have been used, chiefly abroad, as reservoirs of heat to liberate steam when the pressure drops to a certain extent, but to liberate any energy in this way there must be an appreciable pressure drop and to liberate much there must be enormous quantities of water so stored. For large power stations with widely fluctuating loads, requiring momentarily very large quantities of energy, this device has up to the present time proved impractical. Another way in the nature of a possibility of meeting the situation is not to attempt to stamp out the steam-generating variations to meet load variations, but to allow them to exist, and to use a different fuel, such as gas.

At the present time bituminous fuel is used in these furnaces for its gas characteristics, and it is the gas that is depended upon very largely to meet these fluctuations. If the fuel were entirely a gas, then the fluctuations could be met much more easily and there would be no necessity for any sort of energy storage system. Coal can be transformed into gas in the gas producer, and this can be fired in the boiler, but it is a "lean" gas and requires a very large furnace to liberate the same amount of heat that is now liberated on standard grates. Moreover, the present styles of gas producer are large and expensive, and would probably take up more room than the entire power house now does, so that although this is a possible way of avoiding present difficulties, it seems, like the others, impractical. Here is an opportunity for inventors.

A Study of the D Slide Valve

BY E. S. HAWKINS

In January POWER the general features of the Zeuner slide-valve diagram were considered, and it will now be in order to take up more in detail the effects of changing the valve setting, and the methods of using the diagram for designing and proportioning valves for new engines or for remodeling valves in engines already in operation. It will be seen in

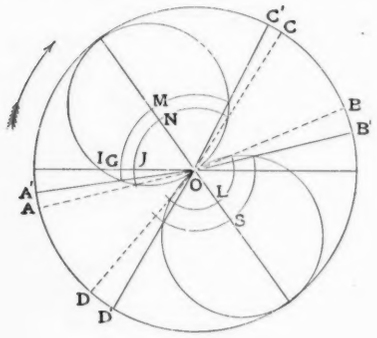


FIG. 1

Fig. 1 that if the outside lap is increased an amount corresponding to N—M on the scale of the diagram, admission will take place later, at the crank position O—A', where the lead will be reduced to I—G, and cut-off will take place earlier at O—C'. On the other hand, the contrary effects are seen where the outside lap is reduced, and if the inside lap be increased an amount equal to L—S release will take place later, or at the crank position O—B', and compression will take place earlier, at O—D', the effect being exactly opposite for a decrease of the inside lap.

In Fig. 2 is shown the effect of altering

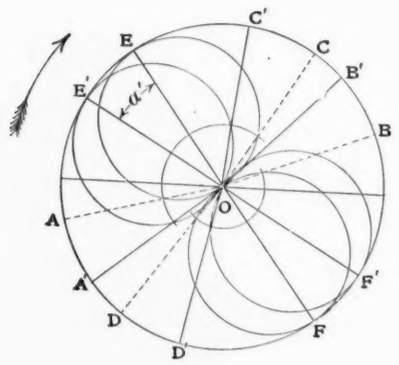


FIG. 2

the angular advance of the eccentric. It will be seen that if the angle of advance is increased, as at a', all the events of the valve occur earlier, since the crank revolves in the direction indicated by the arrow, and the new position of admission O—A' is ahead of O—A, the old position. This is true also of the other valve movements, O—E' being ahead of O—E, and O—C' ahead of O—C.

A third alteration that may be made in the valve gear is a change of eccentricity, by altering the eccentric radius, when making a design, or by using a new eccentric of radius different from that of the older one, where an engine is being rebuilt. The effect of a change in eccentricity is shown in Fig. 3, in which it will be noted that when the eccentricity is increased, the valve travel is increased, and admission takes place earlier (at O—A'), the lead being increased an amount equal to I—I', while cut-off takes place later (at O—C'); O—B' in the figure comes ahead of O—B, showing that release is earlier the greater the eccentricity and, since O—D' comes after O—D, it is plain that compression is later. Since the upper valve circle cuts the arc drawn from O as the center, with a radius equal to the outside lap plus the width of the steam port, in the points W' and H', the admission port will be open wide while the crank moves from O—W' to O—H'. On the lower valve circle it is shown that the exhaust port opens sooner (at W) and remains open longer (at H). It is plain, of course, that decrease of the eccentricity will have a contrary effect.

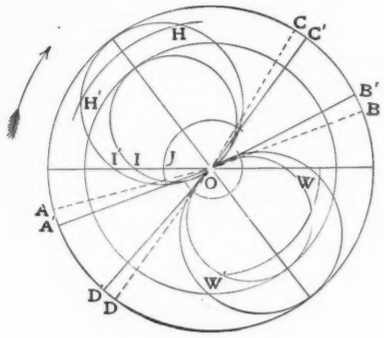


FIG. 3

TABLE SHOWING IN CONDENSED FORM EFFECT OF CHANGES IN LAP, TRAVEL OF VALVE, OR ANGULAR ADVANCE OF ECCENTRIC

	Increasing Outside Lap.	Increasing Inside Lap.	Increasing Travel.	Increasing Angular Advance.
Admission	Is later.	Not changed.	Begins earlier.	Begins earlier.
	Ceases sooner.		Continues longer.	Same period.
Expansion	Is earlier.	Beginning unchanged.	Begins later.	Begins earlier.
	Continues longer.	Continues longer.	Ceases sooner.	Same period.
Exhaust	Unchanged.	Occurs later.	Begins earlier.	Begins earlier.
		Ceases sooner.	Ceases later.	Same period.
Compression	Begins at same point.	Begins sooner.	Begins later.	Begins earlier.
		Continues longer.	Ceases sooner.	Same period.

The accompanying table shows in condensed form the effect of changes in the lap, travel of the valve, or the angular advance of the eccentric.

From what has been said, it will be seen that in all problems using the valve diagram, there are, in addition to the four valve movements of admission, cut-off, release and compression, the other variables of valve travel, angle of lead, outside lap and inside lap. In every problem, some of these items are given and the others must be found and, in designing an engine, the conditions under which it is to be used determine certain of the elements which are then considered fixed, and are used in computations to determine the other variables. In general, the work of any case will fall under one of five problems, each of which will be here explained.

PROBLEM I. Given the travel of valve and points of admission, cut-off and release.

In Fig. 4, let C—P_3 equal the given travel of the valve, and on this line as a

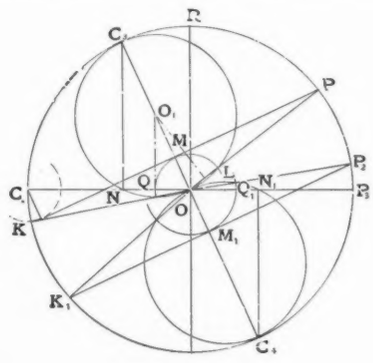


FIG. 4

diameter construct a circle, marking on it the points K, P and P_2 for the given points of admission, cut-off and release. Through K and P draw the line K—P, and through O draw the line C_3—C_4 perpendicular to the line K—P. From P_3 draw the line P_2—K_1 parallel to K—P, and intersecting the line C_3—C_4 at the point M_1, the distance of which from the center O is the exhaust lap of the valve, while the similar distance of the point M on the line K—P from the center O is the lap of the steam valve. By drawing in the valve circles and the lap circles through the points M and M_1, the leads N—Q and N_1—Q_1 are determined.

PROBLEM II. Given the travel of valve, steam lap, steam lead and any point in the exhaust diagram.

In this case the crank circle is constructed as before, and the distance O—N laid off equal to the lap plus the lead. At the point N the perpendicular N—C_3 is erected, cutting the crank circle in the

point C_3. It is plain then that the angle C_3—O—R is the angle of advance and the diagram is completed by drawing the diameter C_5—O—C_4, and following out the construction by drawing in the valve and lap circles.

PROBLEM III. Given the travel of valve, cut-off, steam lead, and exhaust lap.

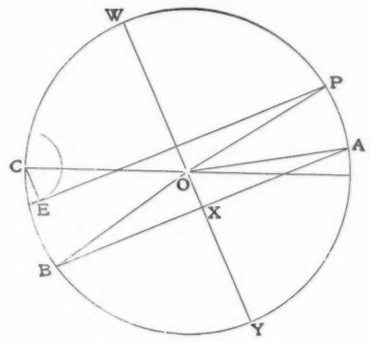


FIG. 5

First construct the valve circle as before and mark on it the point P corresponding to the point of cut-off. With the point C as a center and a radius equal to the lead (which is given), draw in a circle, as shown in the figure, and from the point P draw a line passing tangent to this circle on the under side. This line will cut the large circle at the point K, which is the point of admission. By drawing a diameter through the point O perpendicular to the line K—P the angle of advance is determined, after which the rest of the construction can readily be carried out.

PROBLEM IV. Given the steam lap, steam lead and the point of cut-off.

In Fig. 4, let the angle C—O—P be that corresponding to the point of cut-off and draw in the lap circle $Q\,M\,L$, laying off at the same time the distance Q—N equal to the lead. Connect the points N and O and the points O and L by straight lines, and erect a perpendicular to each of these lines at its middle point. The point O_1 at which these two perpendiculars intersect will be the center of the steam-valve circle and the radius will be the distance from it to the point O, or either of the points N and L. The diameter of this circle is, of course, equal to one-half the travel of the valve, so that the crank circle may readily be constructed, and the angle C_3—O—R between the diameter of the valve circle and the perpendicular erected from the point O will be the angle of advance. It is, then, an easy matter to carry out the construction and finish the diagram by drawing in the other valve circle and the two lap circles, and laying off on the crank circle the points corresponding to the other valve movements.

PROBLEM V. Given the cut-off, release, compression and width of port.

It is assumed that the width of the port is equal to the maximum opening of the exhaust port and to construct the diagram, it will first be necessary to find the travel of the valve, which is done, as shown in Fig. 5, by drawing a circle of indefinite radius and marking on it points of cut-off, release and compression. This can readily be done, since these points if given in terms of the stroke, as explained in the former article, may be found in terms of the angle turned through by the crank. Join the points of release and compression by the line A—B and through O draw a diameter perpendicular to this line. It is then plain that the ratio of valve travel to the given width of port is equal to the ratio of W—Y to X—Y; and from this the travel of the valve is obtained and used as the diameter of a new crank circle, which, when drawn, has the points of cut-off, release and compression laid down and the diagram drawn as per the problem already given.

Sometimes the point of release is given instead of the width of port, and in this case the travel of valve is found by drawing through the point P a line parallel to A—B and from the point C dropping a perpendicular to this line at E. The line C—E is then proportional to the lead so that we have the ratio travel of valve to given lead equals W—Y to C—E, from which the travel of valve is found as before. In these cases the distance W—Y is multiplied by the width of port or by the given lead, as the case may be, and the product divided by the distance X—Y or C—E, to find the travel of the valve in actual inches. This dimension must, of course, be altered to correspond with the scale selected for drawing the diagram.

As illustrations of the practical working of the diagram a number of examples with solutions will be given.

1. Travel of the valve is 6 inches and cut-off occurs when the crank has completed 105 degrees of its path. Admission of steam begins when the crank is within 7.5 degrees of the beginning of its stroke, and exhaust closes when it is 60 degrees from the end of its stroke. Construct the Zeuner's valve diagram and find the steam lap, steam lead, exhaust lap, exhaust lead and the angle of advance.

As the first step, select some scale to which the diagram is to be drawn and, as in this case the travel is 6 inches, one-half this length may conveniently be used for the diameter of the crank circle, which is drawn as shown in Fig. 6. On this circle, the given points of admission, cut-off and compression are laid down at K, P and P_4, respectively, making the arcs C—K, C—P and C—P_4 equal, respectively, to the given angles of 7.5, 105, and 60 degrees. A line is then drawn connecting the points K and P, and, through the center O of the circle, a diameter is drawn perpendicular to this line, intersecting the

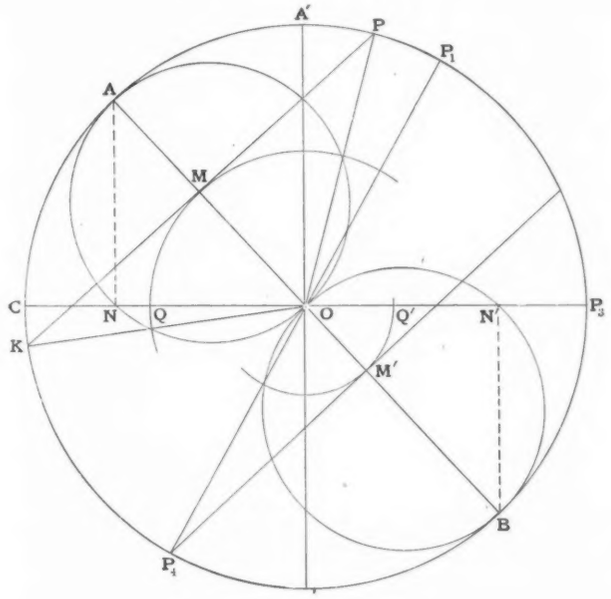


FIG. 6

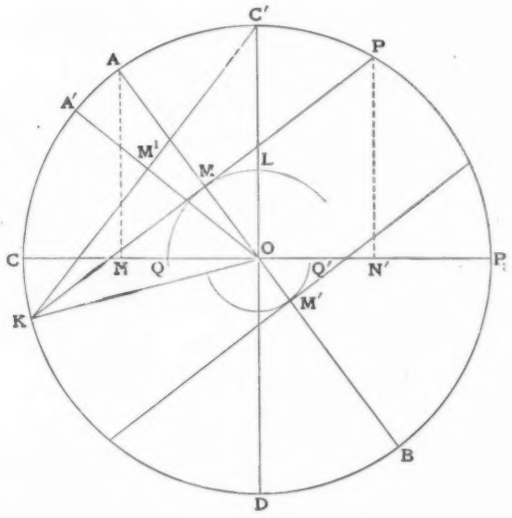


FIG. 7

crank circle in the points A and B and the line K—P at M. By measurement, the angle A—O—A' is found to be 42.5 degrees and this is the angle of advance required to be found. With the lines O—A and O—B as diameters, valve circles are drawn in and, with O as the center and O—M as the radius, the steam-lap circle is drawn, O—M being one-half the steam lap, which by measurement is found to be 13/16 of an inch. The distance N—Q between the intersections of the valve and the lap circles with the line O—C is found to be 3/16 of an inch, and this is one-half the steam lead.

To find the exhaust lap and lead, draw from the point P_4 a line perpendicular to the diameter A—B, and intersecting this diameter at the point M', the distance of which from the center O of the crank circle is the exhaust lap, so that the circle described with O as its center and the distance O—M' as a radius is the exhaust-lap circle and the distance Q'—N' between its intersection with the line O—P_3 and that of the exhaust-valve circle is found to be 9/16 of an inch, which is one-half the exhaust lead of the valve; O—M' is 7/16 inch, so that the exhaust lap is 7/8 of an inch.

Here the travel of the valve is 5 inches and one-half of this may be taken as the diameter of the crank circle, which is shown in Fig. 7 with the two diameters C—P_3 and C'—D at right angles. O—M O as the center and the radius O—M equal to one-half of the given steam lap of 1 inch, the steam-lap circle Q M L is drawn in, and with a radius of 1/4 of an inch the lap circle for the exhaust valve

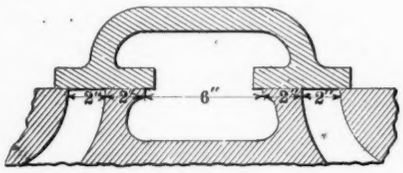


FIG. 8

is also drawn. From the point Q, where the steam-lap circle intersects the horizontal diameter C—P_3, the distance Q—N, equal to 1/4 of an inch, or one-half the given steam lead, is laid off and, from the point N, a perpendicular is erected on the diameter C—P_3, intersecting the crank circle at the point A, from which the diameter A—O—B is drawn.

The angle A—O—C' is the angle of advance and by measurement is found to be 37 degrees. Since the connecting-rod length is considered infinite, the distance of the piston from the end of its stroke at cut-off will be equal to the distance N'—P_3 from the foot of the perpendicular dropped on the diameter C—P_3 from the point of cut-off P. By measurement this distance is found to be 5/8 of an inch, and since the scale of the diagram is 2½ inches equal to the stroke of 4 feet, the distance of 5/8 of an inch will be exactly one-fourth stroke, so that this is the point of cut-off. This point is found on the diagram by drawing the line K—P through the intersection of the steam-lap circle with the diameter A—B at the point M, making the line K—P perpendicular to A—B. In this way, also, the admission point K is found.

To find the amount of lap that would be required to make the cut-off come at half stroke, assume the cut-off to be at this point, or at the line O—C' in the figure, and from C' draw a line to the point of admission at K; O—A' is then drawn perpendicular to K—C' and the distance O—M_1, which by measurement is found to be 3/4 of an inch, is one-half the steam lap that would be required to bring cut-off at half stroke. With this condition, the angle of advance will be A'—O—C', which by measurement is found to be 51 degrees. This greater angle of advance would make the other movements of the valve come earlier in the stroke.

3. A valve such as shown in Fig. 8 is to have a travel just sufficient to open the port wide for exhaust, when the exhaust is a maximum, and with a connecting-rod considered of infinite length it is to cut off at three-fourths of the stroke. The exhaust port is to begin opening when the piston is at one-eighth of its stroke from the end and admission occurs at one-sixteenth of the stroke. It is required to find the steam and exhaust laps, the travel of the valve, the angle of advance of the eccentric and the position of the piston at exhaust closure.

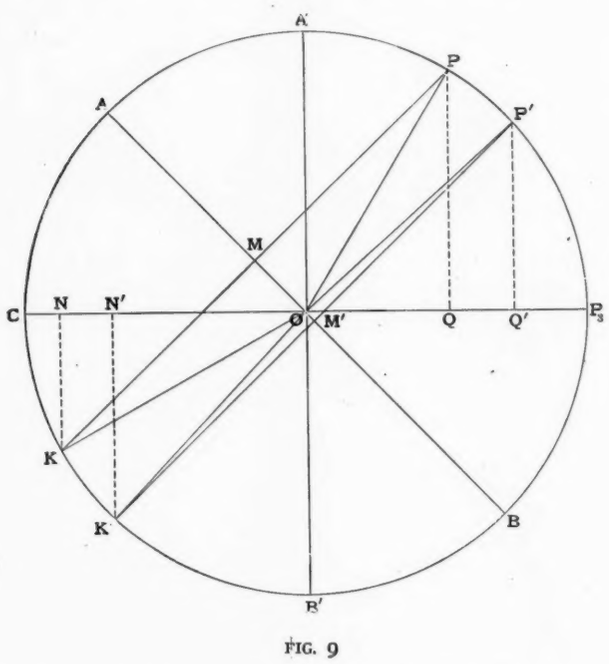


FIG. 9

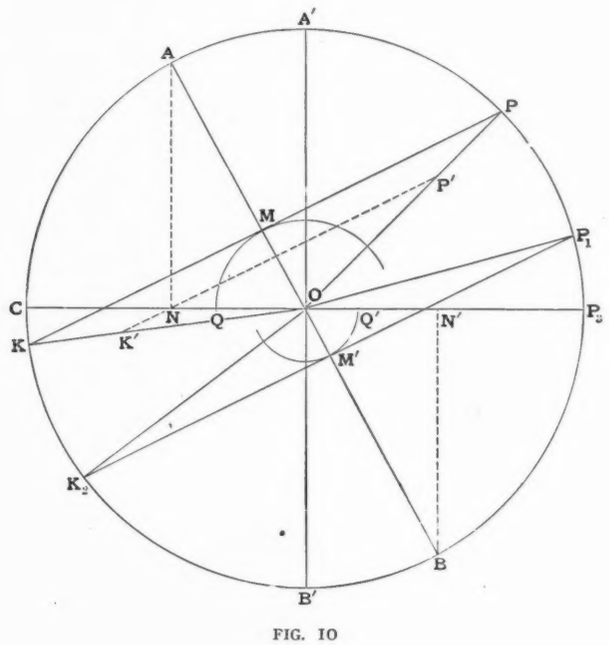


FIG. 10

2. A valve has a travel of 5 inches, a steam lap of 1 inch, an exhaust lap of 1/2 inch, and a steam lead of 1/2 inch. It is required to find the angle of advance and the point of cut-off. The stroke of the engine is 4 feet and the connecting-rod is considered to be of infinite length, so that its angularity may be disregarded. It is also required to determine what should be the steam lap of the valve if steam is to be cut off at half stroke; what should be the angle of advance to make the engine cut off at half stroke, and how this change would affect the other functions of the valve.

To solve this problem draw a circle, as in Fig. 9, making the diameter of any convenient length, which in this case is taken as 3 inches, and draw also the diameters C—P_3 and A'—B' at right angles. On the diameter C—P_3 the distance C—Q is 2¼ inches, or three-quarters of the diameter. At this point erect a perpendicular to the diameter, intersecting the circle at the point P, which will be the point of

cut-off. In a similar way the point Q' is located, so that its distance from the point P_3 is 3/8 of an inch, or 1/8-stroke from the outer dead center P_3, and the perpendicular erected from this point Q' intersecting the circle at the point P' locates the point of release. In a similar way the point N is found 1/8 of the stroke, or 3/16 of an inch from the point C and the perpendicular dropped to the circle, intersecting it at K, the point of admission.

By connecting the points K and P with line $K—M—P$ and drawing a diameter through the center O perpendicular to this line, the angle of advance $A—O—A'$ is located and this is found to be 44 degrees. To find the point of exhaust closure or compression, draw from P' a line perpendicular to the diameter $A—B$ and intersecting the crank circle at the point K' which will be the point of compression. By drawing from this point a line perpendicular to the diameter $C—P_3$ and intersecting it at the point N', which is 1/2 inch from C, it is seen that compression takes place when the piston is 1/6 of its return stroke from the inner dead center C. Fig. 8 shows that the width of the port is 2 inches and this, therefore, is the maximum opening to exhaust, so that, according to the conditions given in Problem V, the travel of the valve ÷ given width of port = $A—B \div M'—B$, Fig. 9. The width of the port is 2 inches, as just stated, so that the travel of the valve is equal to $2 \times 1\frac{1}{2} \div M'—B$, which is found by measurement to be 1 13/32 inches. This gives the valve travel as 2.13 inches (4.26 inches actual, as the diagram is half scale) so that all linear dimensions taken from the diagram as drawn in Fig. 9 must be reduced in the ratio of 2.13 to 3. Reducing the distance $O—M$, which measures 3/8 of an inch by this ratio, we find that the actual steam lap is 0.532 inch; and the exhaust lap $O—M'$, reduced in a similar way from its actual measurement of 3/32 of an inch, gives the actual amount of lap as 0.133 inch.

4. In a given engine, cut-off is to take place when the crank is 45 degrees from the end of its stroke, release being at 15 degrees from the end of the stroke and admission at 7.5 degrees from the beginning of the stroke. The maximum exhaust port opening is 2.5 inches and it is required to find the travel of the valve, the steam and exhaust laps, the crank angle when compression begins and the exhaust lead.

Again making the diagram one-half full size, draw the two lines $C—P_3$ and $A'—B'$ perpendicular to each other, as in Fig. 10, and from their point of intersection O draw the line $O—K$, making an angle of 7.5 degrees with $O—C$, also the line $O—P$, making an angle of 45 degrees with $O—P_3$ and the line $O—P_1$, making an angle of 15 degrees with $O—P_3$. We then lay off from the point O on the lines $O—K$ and $O—P$ equal distances, 1½ inches, as shown in the figure, and connect the points K and P thus found. Through the point O the line $A—B$ is drawn perpendicular to the line $K—P$ and from the point P', which is also 1½ inches distant from O, the line $P'—K_2$ is drawn parallel to $K—P$, so that it is also perpendicular to the line $A—B$. From its point of intersection M' with the line $A—B$, lay off the distance 1¼ inches to the point B. With O as the center, and with $O—B$ as a radius, draw the circle as shown. This will be the crank circle and its diameter $C—P_3$ the travel of the valve, which by measurement is found to be 3 inches (6 inches actual) while the distance $O—M$, or the steam lap, is by measurement found to be ½ inch (1 inch actual) and the exhaust lap $O—M'$, ¼ of an inch (½ inch actual), the crank angle at compression $C—O—K_2$, 36 degrees, and the steam lead $N—Q$, ¼ inch on the diagram or 0.5 inch actual, while the exhaust lead $N'—Q'$ is 13/32 of an inch on the diagram or 13/16 actual.

It so happens in this case that the distance selected to lay off on the lines $O—P$ and $O—K$ was 1½ inches, or one-half the valve travel, but this is not necessarily the case, as the solution will be unaffected no matter what distance is laid off. Thus, for instance, if we should lay off 1 inch on $O—P$ and $O—K$ to the points K' and P' and join these points by a line as shown, we would still be able to draw the line $A—B$ perpendicular to the dotted line so that its angle with the vertical $A'—B'$ would be the same as at first. All of the linear dimensions of the diagram are determined by the maximum opening to exhaust, 2½ inches, which is laid off from the point M' to B, as already described, for determining the radius for the crank circle.

An Emergency Change-over in a Large Turbine Station

BY E. RUSSELL

The following incident occurred in a large traction power station, near London, England, equipped with double-flow turbines of the most modern type, having a capacity of fourteen thousand kilowatts.

A new machine had been started up a few days previously for the purpose of drying out the generator, and it having been pronounced fit for service, it was decided to run it in parallel with the old unit after the heavy evening peak was over, until the track load was well under full limit for a single machine. The other machine was then to be shut down, leaving the new one to complete the day's service.

Shortly after 11 p. m. the first machine was cut out and all went well until just before midnight, when a fairly heavy load came on in consequence of empty trains being brought home and an experimental one running out to test some new track.

The first inkling received of trouble was when there was an apparent earthquake under the turbine, accompanied by a racket as if a dozen men were filing the edges of thin steel plates. The glands were inspected, but there was nothing wrong with them, and as the noise continued, some of us were for immediately shutting down. The man in charge of the turbines would not hear of dropping the load through them, so another set was hurridly got up to speed and put in parallel. The affected machine was stopped and the equalizing pipes taken down to see if any blades or pieces had got into them (as in cases of strips at one end pieces of blade are often found in these, being left there by the steam in flowing to the other side to equalize the pressure), but they were found to be quite clear. Next, the relief valves at each end of the machine were taken off, but still all was O. K., so it was decided to lift the cover, which weighed about twenty tons. On this being done, nothing appeared wrong at first, but when the rotor was turned around, it was discovered that not only had the impulse blades on the rotor rubbed slightly on the cylinder in one or two places for a distance of a few inches, but, that the real mischief was in the fixed row of guide impulse blades. These had evidently not been ground out truly, nor had they been given the correct clearance, and to make matters worse the spindle gave signs of not being exactly round, with the result that a section about three feet long had been badly galled by the shrouding of the impulse guide blades. Of course, this shrouding had not come off "scott free," so after a lot of filing and allowing greater clearances the unit was once more assembled and run. None of the Parsons blades were hurt in the least, and the apparatus seemed little the worse for the shaking.

Fom the time this machine first started to show signs of trouble until the second had been run up and paralleled, and the former shut down, a period of three minutes had elapsed. The air pump was kept running: thus, only the circulating pump had to be got away, and the oil turned on the bearings. The condensers in this case are of the barometric type with the atmospheric exhaust on the top of them, so that it was impossible to start up on to the atmosphere, and then change over to vacuum. The machines are rated at thirty-five hundred kilowatts, and run at one thousand revoluions, generating at eleven thousand volts pressure.

It having been learned that the climate, soil and other conditions of the Philippine Islands are favorable to rubber and gutta percha cultivation, the Government has established nurseries in the islands for the distribution of rubber plants to plantations.

Hints on the Practical Management of the Storage Battery—Precautions Necessary in Charging and Discharging End Cells

BY WILLIAM KAVANAGH

Usually the number of cells in a storage battery exceeds the required amount by ten or twelve, the object in having more cells than is needed to give the desired voltage being explained as follows: Suppose it takes 56 cells to give 112 volts at a given amperage or outflow of current; then, if there are no "end cells," or in other words "reserve cells" to cut into the circuit when the voltage begins to fall, the battery will run down and the lights become dim. Likewise, if motors are running they will slow down and eventually stop; hence the reserve or end cells are kept ready to be cut in on the circuit at the proper time, to increase the voltage and maintain it practically constant. In general, the reserve cells are not cut into the circuit unless the voltage falls below a predetermined point, and if the number of end cells in use is sufficient to maintain the required voltage up to the time the engine is started, or charging begins, the remaining end cells are not charged.

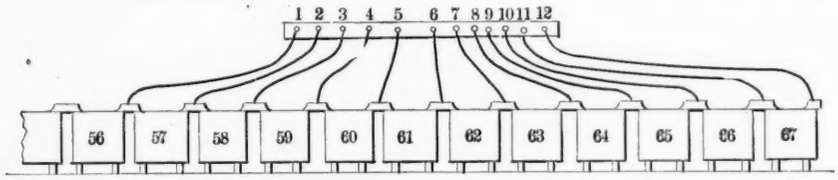


FIG. 1. SIDE VIEW OF TWELVE END CELLS AND CONNECTIONS

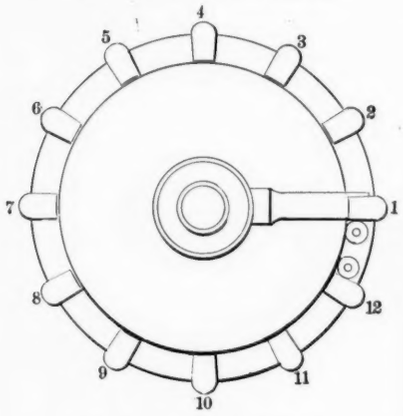


FIG. 2. END-CELL SWITCH

Fig. 1 is a side view of twelve end cells and their connections. The numbers 1 to 12 coincide with the numbers 1 to 12 on the end-cell switch, shown in Fig. 2. If No. 1 connection is taken from the fifty-sixth cell, and the end-cell switch is on this connection, then if the voltage has fallen only one or two volts up to the time of starting the engine, it will be only necessary to charge the battery from cell 56 down. In practice, however, this condition is seldom attained unless very little current is being taken out of the battery.

In most plants where storage batteries are employed, probably one-half, or it may be the whole number, of end cells will be connected to the bus-bars before the generators are started. In this case all of the end cells must be charged. The booster panel will contain an end-cell switch similar to that in Fig. 2. Referring to the illustration, it will be seen that the switch is cut in on No. 1, which connects with cell 56, and should the voltage on that circuit fall the switch may be shifted to No. 2, which connects with cell 57, and so on around the switch until finally No. 12 is reached. Cell No. 12 is the last reserve; consequently the generator must be started before the voltage begins to drop too rapidly. Should the end-cell switch be cut in on No. 6 (which connects with cell 61) at the time the engine starts, it will not be necessary to move the switch around to 12 in order to begin charging; but this is with the proviso that the remaining end cells are in good condition, and are not in need of being charged.

There are three approved ways by which to tell when charging should begin: First, by the lowering of the voltage, requiring the use of all or nearly all of the end cells; second, by the specific gravity of the electrolite; third, by the change in color of the positive plates, and by gassing in the cells having nearly ceased. When discharging, the positive plates gradually change from the normal chocolate until they nearly approach the color of the negative plates.

The specific gravity of the electrolyte is determined by the hydrometer, and as the specific gravity changes under charge or discharge, the hydrometer is a very reliable instrument to employ in determining when to begin and when to cease charging. The compensating type of hydrometer is generally used. In operation, the specific gravity of the cells usually ranges between 1185 and 1195. This is ten points and it should be noted, first, that the gravity should not be allowed to fall below 1182, nor rise above 1197; second, the charge should begin immediately after discharge, or as soon as possible after the batteries are cut out of circuit; third, the batteries should not be overcharged oftener than once in each week. When the overcharge is complete, the hydrometer will cease to indicate further increase in specific gravity.

The primary function of the booster is to elevate the voltage above that of the line voltage, in order that the cells may be charged. The booster panel board shown in Fig. 3 is fitted with the following instruments: Voltmeter *V;* recording ammeter *A;* booster starting switch *B;* over- and under-load quick-break switches *M* and *N;* single-pole switch *S;* end-cell switch *B'*; multipolar voltmeter switch *Y'*; charge switch *C;* discharge switch *C'*, and a booster rheostat *R.*

To charge the batteries, raise the voltage on the generator to equal the voltage indicated by the battery voltmeter and throw in the main switch of the generator. Next pull out the discharge switch of the battery, and start the booster by means of the switch shown at *B.* After the booster is up to speed, close the single-pole switch *S* and note the voltage; when the generator voltage equals the battery voltage, close the charging switch and cut out the resistance at *R* until the voltmeter shows two volts for each cell. Thus, if there are 60 cells to charge, the voltmeter should indicate 120 volts. This rate of charge will not remain fixed, and in order to maintain it constant or approximately so, as the battery voltage rises, more resistance must be cut out at *R,* and when all the resistance is cut out, it will help matters to shift the generator brushes on the booster.

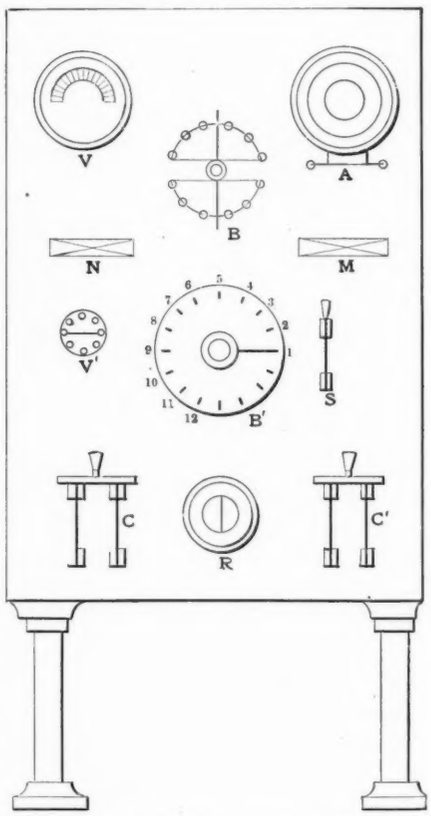


FIG. 3. BOOSTER PANEL-BOARD

If at the beginning of the charge the cell switch is on contact number 4 (cell 59), leave the switch on this cell and charge. At the expiration of one hour shift the switch to No. 3, and at the end of the second hour shift the switch to No. 2, when cell 56 will receive the last charge and the switch should remain on this cell

connection as long as the battery maintains the voltage.

With respect to the length of time required to properly charge each end cell, no rule can be given, but the charge should be divided up between the cells in such a way that when the first cell is reached (in this case 56), the heaviest of the charge will not have to be delivered through this cell. Thus, if the gravity of the battery falls 10 points during discharge, and five end cells are in use, and it takes five hours to charge, each end cell may be charged for one hour. In this way each cell will receive a proportionate amount of the charge, and be therefore capable of adding voltage when required.

As there are several ways of telling when charging should begin, so there are different methods by which to know when the batteries are charged. First, by the hydrometer; second, by the color of the positive plates, and third, by the gassing of the cells. The specific gravity of the cells will rise as the amount of charge approaches the normal limit, and by inspection of the voltage, the rate and time of discharge, and the number of end cells in use, the limit accorded the specific gravity both for charge and discharge can be determined.

The proper care of the cells is just as important as their charging and discharging. Since gassing will lessen the amount of electrolyte in each cell, it follows that part of the surface of the plates will become exposed. To obviate this, sufficient water should be added when charging begins, or at the end of the discharge, to submerge the plates wherever exposed. The water should never be added at the end of the charge. It is important to test each cell for voltage at least once every two weeks, a portable voltmeter being used for this purpose.

Sediment accumulates beneath the plates and must be removed, but a metallic tool should never be used for this purpose, nor to clean between the plates. A good plan is to lower an incandescent lamp into each cell, and the amount of sediment and the condition of the plates will then be visible. The lamp should be made waterproof by running the wires in a vulcanized rubber tube, and by cementing and taping the lamp end of the tube, the other end being fitted with a socket connected to any convenient point by means of flexible lamp cord.

The gravity of each cell is also important, and when adding acid to the batteries the gravity of the cells should be kept as nearly equal as possible. The cells lowest in gravity can be strengthened by adding diluted acid and in this way the gravity of the cells can be made nearly equal. When adding acid or cleaning the cells woolen clothing should always be worn, as the acid will inevitably destroy anything in the shape of cotton.

Two Narrow Escapes—Disasters Averted by Timely Discovery of Loose Nuts

BY W. H. WAKEMAN

There are illustrated and explained in the mechanical papers from month to month so many engine wrecks in which lives are lost and much valuable property destroyed, that it appears as if an engine is almost as dangerous as a boiler at the present time.

These illustrations are object lessons which should lead to precautions that will prevent them elsewhere, but they do not always answer this purpose. Two cases are presented herewith which show narrow escapes from serious accidents, and as other engines now in service may be found in the same condition, if a remedy is promptly applied in every such case it will save future trouble and expense.

A certain engine pounded badly on the head end occasionally, but never on the crank end. This is one of the worst kinds of pounds to locate, as it may not be possible to find it when heard, and when a search is fairly begun it disappears as if by magic.

I made an examination of the external parts, but found nothing that served to locate the pound. The cylinder-head was removed and on putting a socket wrench on one of the follower bolts to remove it, the whole piston and rod turned together. This showed at once that the piston-rod was loose in the crosshead, but when the piston was turned back to the position in which I found it, there was no evidence to show where trouble was to be found, because the check-nut was apparently in its place, although not duly fastened.

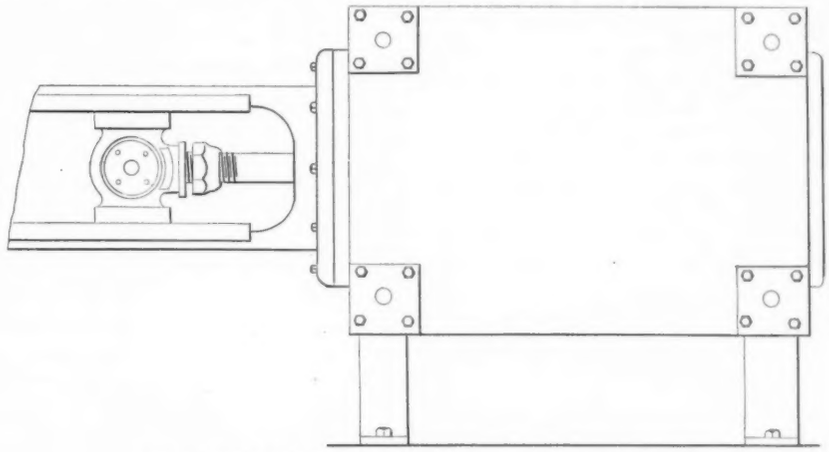


FIG. 1. HOW LOOSE PISTON-ROD TURNED IN CROSSHEAD

This is what took place in daily practice. When turned to the position in which I found it there was little or no noise, but occasionally the rod turned out of the crosshead, as shown in Fig. 1, until piston and cylinder head came together, causing a heavy pound on the head end at each revolution. Had this turned much farther it must have caused a bad wreck, but it would turn in again, stopping all noise for several hours, after which the trouble would be repeated, and this had continued for several weeks. Of course it did not take long to put the piston in place and tighten the check-nut with a suitable wrench, but it was a narrow escape from serious trouble.

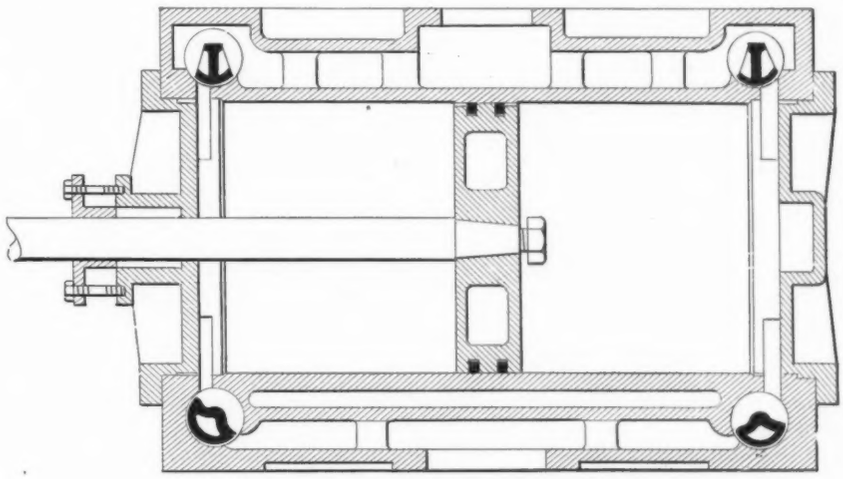


FIG. 2. HOW LOOSE NUT FORCED THE PISTON OFF

Fig. 2 illustrates another case in which

there was a heavy pound once in each revolution, but here it was on the crank end. This was a larger engine than previously mentioned, as it developed more than 1000 horse-power; therefore, every time it was shut down and its internal parts examined it cost several dollars.

The connecting-rod is a tapering fit in the piston which is held in place by a large nut. This nut became loosened, hence when steam was admitted to the crank end of the cylinder it forced the piston off until the nut prevented further movement at this point. This jarred the whole engine and caused it to vibrate badly, but when the head end was reached and a charge of steam admitted to this end of the cylinder, it forced the piston on again, as shown, but on account of the tapering rod this was a comparatively easy movement, making little noise.

Of course this was a very dangerous condition of affairs, but the engine was run several days after efforts had been made to locate the cause of trouble, until an expert was summoned, discovered the loose nut, and tightened it in about five minutes.

He was paid well for his services, and this, together with the cost of other efforts made to remedy the evil, resulted in a total expense of not less than $500. Had the nut come entirely off, it might have proved much more expensive.

These two incidents should prove beneficial to owners of steam plants and engineers that have troubles of this kind, for "history repeats itself," and what has happened in one plant may happen in any other of similar design.

Induced Draft in the Boiler House

In a paper by an English author, on "Notes on the Application of Induced Draft," the author referred to one case in particular of five boilers fitted with steam jets, which resulted in a coal consumption of 27.3 pounds of coal per square foot of grate, the evaporation from and at 212 degrees being 9.8 pounds of water per pound of coal, with a temperature of the economizer water of 166 degrees. With induced draft the results obtained were as follows: Coal consumption per square foot of grate, 25.4 pounds; evaporation of water, 10.2 pounds per pound of coal; and temperature of economizer water, 259 degrees. Taking measurements of the electrical output of the plant, it was found that, using the steam jets, for each unit of electricity 3.6 pounds of coal and 30 pounds of water were consumed. With induced draft the results were 3.1 pounds of coal and 28 pounds of water per unit of current. In this case 1800 gallons of water were simply evaporated for use by the steam jets. The net saving in this case worked out at 14½ per cent.—*Engineering.*

THE HYDRAULIC ELEVATOR—IV

Simple Low-pressure Vertical Type with Hand-rope Control

BY WM. BAXTER, Jr.

The fundamental principles of the different types of hydraulic elevators having been discussed in the preceding articles of this series, the constructional features of each type will next be considered. This article is devoted to the simplest form of vertical low-pressure elevator, such as is installed in buildings from five to seven stories high. An elevator of this class is very clearly represented in Fig. 25, which shows an Otis machine geared two-to-one. Looking at this illustration it will be seen that the general arrangement is the same as that of the diagram Fig. 3 in the December number, and from this fact it might be inferred that the elevator is not counterbalanced. This, however, is not the case; a counterbalance is used, but it is placed within the cylinder, resting on top of the piston. This construction is very common with two-to-one machines, and even with higher gears. Generally a portion of the counterbalance is placed on top of the piston, so that in such machines the counterbalance weight is divided into three parts, one being within the cylinder, one in the traveling sheave frame, and one constituting the independent counterbalance.

It will be noticed in Fig. 25 that there are two piston-rods *R*. This construction was adopted in the early days of hydraulic elevators partially to increase the safety of the apparatus, but principally to prevent the traveling sheave *B* from twisting around. The ropes tend to hold the sheave from twisting, but they will not prevent slight movements, while the double piston-rods will. Now and for several years past, however, the frame of the traveling sheave has been made in the form of a crosshead running in stationary guides, thus effectually preventing any side movement of the sheave. With this construction the main benefit of the double piston-rods is additional safety; while it is possible for one rod to break or become loose, it is practically impossible for both to give way at the same time.

The arrangement of the cylinder *C*, the circulating pipe *K* and the valve *V*, in Fig. 25, is the same as in the diagram Fig. 7, even the inlet *I* being similarly situated. The small pipe *c* is for the purpose of carrying off the drip from the upper side of the top cylinder head, ordinarily, and also for the purpose of draining the water from the upper end of the cylinder, in cases where it is necessary to run the piston to the top of the cylinder to renew or adjust the packing. Some cylinders are arranged to be packed from the upper end and others from the lower end, the latter design being the one generally used in modern machines. As will be noticed, the pipe *c* connects at the bottom of the cylinder with other pipes that connect to the valve chest and the lower end of the

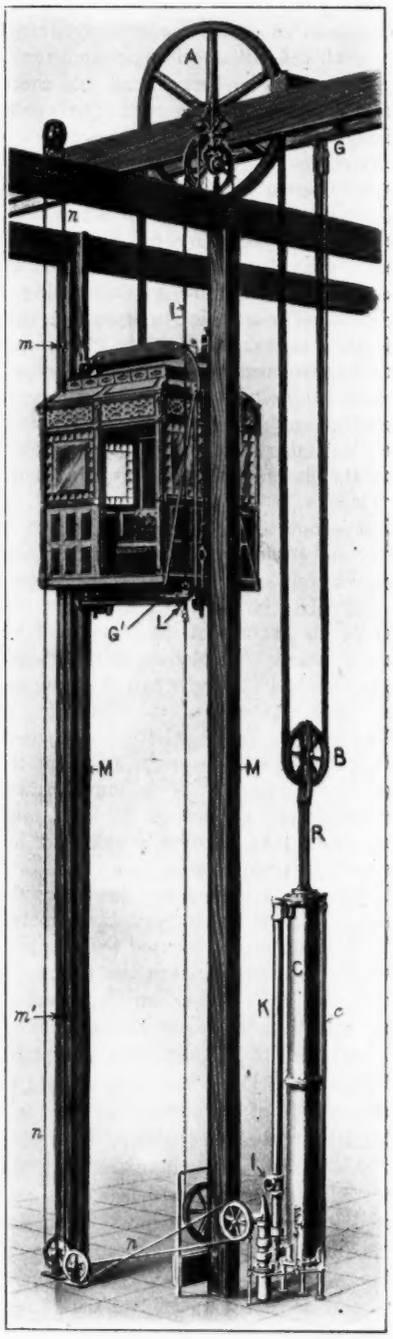


FIG. 25

cylinder. All these pipes are either to carry off the drip or to draw water from the various parts of the cylinder and valve chest when desired. Globe valves are placed in the drainage pipes so as to keep them closed normally.

The elevator is operated by the move-

ment of the hand rope *n*, which passes around a sheave, at the side of the valve chest, which moves the valve through a rack and pinion gear, thence under two small sheaves at the bottom of the elevator well, and from there upward to the top of the well over another small sheave.

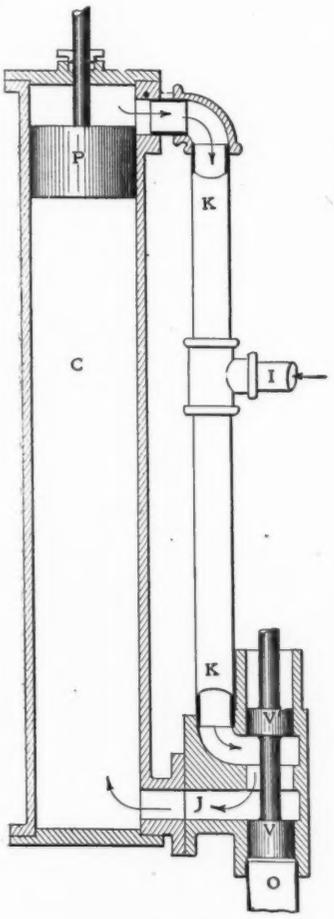


FIG. 7

One side of the hand rope passes through the elevator car, and by pulling this side up the operator causes the car to descend, while by pulling it down he causes the car to ascend. It will be noticed that near the top and bottom of the well balls *m* and *m'* are placed upon the hand rope. These balls are made of such a size that they cannot pass through the openings in the floor and roof of the car through which the rope passes; therefore, when the car, running upward, strikes the upper ball *m*, the latter goes up with the car and pulls upward the hand rope, thereby moving the control valve back to the stop position. Should the car fail to stop, the valve would be carried beyond the stop position and would connect both ends of the cylinder so as to cause the car to run down. This reversal of the motion of the car cannot occur when everything is in proper adjustment, for under such conditions, when the valve is completely closed, the car will stop. If, however, the car should run away by any mishap, it might run beyond the normal limit of travel, and then the valve would be slightly opened in the opposite direction, just enough to develop a retarding force sufficient to stop the car. The action when the car approaches the bottom floor is the same as when it approaches the top; that is, the lower stop ball *m'* is struck and carried down with the car, thereby closing the operating valve. The balls *m m'* are known as automatic top and bottom limit stops, and constitute one of the most valuable safety devices with which elevators are provided, although this fact is not generally realized as fully as it should be. Most men appear to regard them as convenient devices used to stop the car if the operator fails to do so, and to think that if they were not used, the car would simply strike the bumpers a rather hard blow. This would be the case with very slow running cars, but at high speeds, if the automatic limit stops were not used, serious results would be produced if the operator neglected to stop the car in time.

It can be readily seen that if the capacity of the lifting cylinder is sufficient, the car can be run at a much higher speed than is desirable if the valve is opened to its full capacity. To obviate this difficulty, stops are provided so that the operator when pulling on the hand rope cannot open the valve beyond the amount necessary to give the desired car speed. These stops, which are usually mounted at some convenient point in the elevator well, are set above and below the stop balls *m m'*, so as to limit the distance through which the latter can be moved. In some cases additional stop balls are used, on account of its not being convenient to place stops to act directly upon *m* and *m'*. The positions of these stops that limit the amount of opening of the valve are determined experimentally when the elevator is installed.

To keep the car from striking the sides of the elevator well it is run between guides, shown at *M M* in Fig. 25. In the construction here illustrated the guides are made of hard wood. The car is guided at the top by shoes that fit freely against the guides and are provided with means for adjusting them so as to be neither too tight nor too loose. At the bottom the car is guided by jaws formed in a safety device that was formerly known as a safety plank, but at the present time is generally spoken of simply as a "safety." It received the name of "safety plank" from the fact that it is made of a massive hard-wood plank, varying from 4 inches thick and 11 inches wide in the smaller sizes, to 5 inches thick and 15 inches wide in the larger ones.

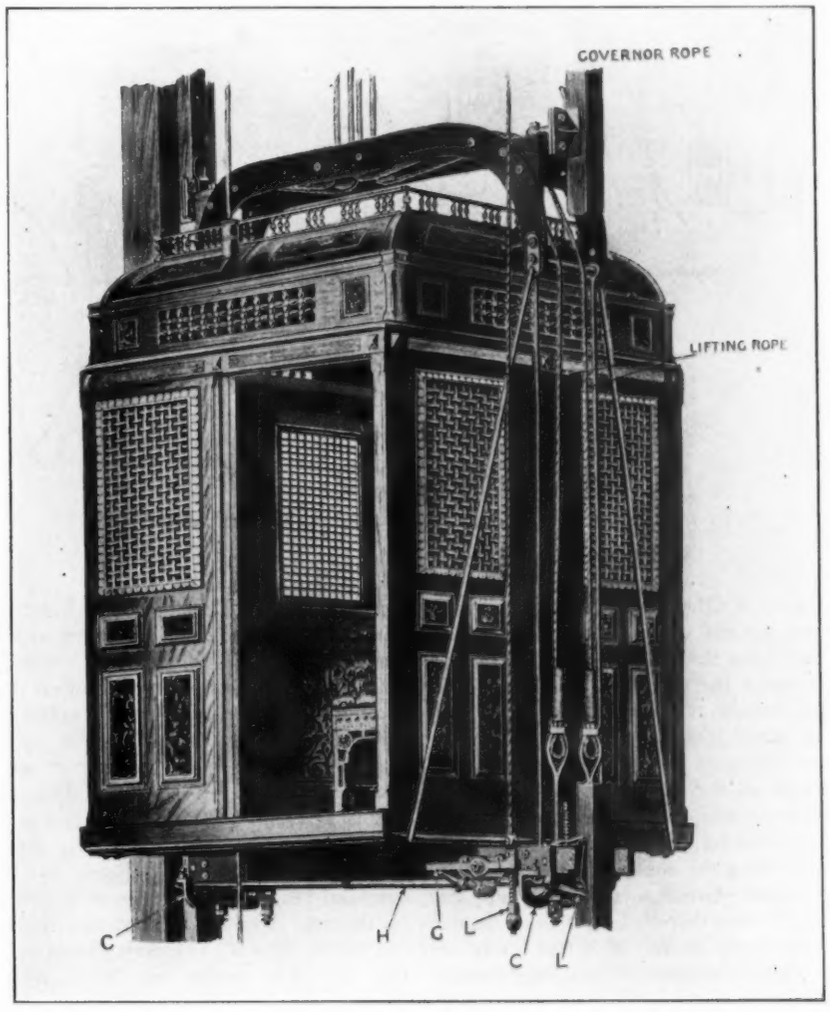


FIG. 26

The jaws of this safety are reinforced with massive iron castings, and on one side are provided with a wedge that can be adjusted in position by means of screws, and on the opposite side with another wedge that can be forced between the guide and the jaw to stop the car if one of the lifting ropes breaks, or the car attains an excessive velocity from any cause. This safety is not very clearly shown in Fig. 25. By the aid of Fig. 26, and the drawing Fig. 27, which shows one end of the safety plank, its construction and operation can be fully understood. In the latter figure the governor rope rod L is shown only in the end elevation.

Looking at Fig. 26 it will be seen that the two lifting ropes that run down to either side of the car are connected with the ends of a rocking lever C. This lever C, as shown in Fig. 27, is pivoted at D', hence if either one of the lifting ropes breaks, the end of the lever it is attached to will drop down. The shaft H, which runs under the car from one side to the other, carries at its ends a lever L' that when raised lifts the wedge N and forces it into the space between the guide M and the side of the jaw of the safety plank. Whichever way the lever C may be tilted by the breaking of one of the lifting ropes, it will rotate the shaft H and lever L' in the proper direction to throw up the wedges N and thereby lock the car against the stationary guides M. The levers on the end of the shaft H are long enough to strike the guides M, when raised high enough, and are sharp at the ends so that they will cut into the guides.

It might be thought that if the wedge N is only raised far enough to catch in the space between the guide M and the safety-plank jaw it would be forced upward so tightly as to stop the car without further assistance. This would be the case if the wedge had a sufficiently long taper, but if it were so proportioned, it would require an enormously strong jaw to resist the bursting strain; moreover, the car would be so tightly wedged that it would require a greater force to release it than could be easily obtained. With the wedges of the proportions used, it is necessary to make the lever that lifts the wedge so that it will dig into the guide, and as the car moves down through, say, a foot or two in coming to a stop, the lever shaves the side of the guide, thereby not only forcing the wedge tighter against the guide but producing an additional retarding force. When a car is caught by the safety, all that is necessary to release it is to start in the upward direction, and the force exerted by the lifting cylinder is enough to overcome the friction of the wedges against the guides.

In the foregoing we have shown how this safety acts providing one of the ropes breaks. Elevator cars, however, seldom drop when one of the ropes breaks, but frequently attain very high velocity when the ropes do not break, and on that account it is necessary that the safety be arranged so as to act when the speed reaches a certain point, no matter what causes the increased velocity. This result is accomplished in the safety shown in Fig. 25 by means of the Otis safety governor seen mounted on one of the overhead beams. This governor is driven by the rope L which is fastened to one end of the lever G', as clearly shown at G, Fig. 26. The spring that holds G' is strong enough to keep the lever in the normal position and rotate the safety governor; hence, the latter will rotate at a velocity proportional to the speed of the car. A drawing of the governor is shown in Fig. 28, and it will be seen from this that the governor can be adjusted by means of the spring on the spindle to act at any desired velocity. The governor driving rope passes through the clamping jaws $H\,H'$, and when the governor speed becomes great enough to lift the rod Z and throw the jaws together, the rope will be clamped. Then, as the rope cannot move, the outer end of the lever G' on the safety plank will be held stationary as the car descends; hence, the shaft H will be rotated, throwing the safety wedges N into action to stop the car.

It is evident that the car can descend only as far as the upward movement of the end of the lever G' and the compression of the spring on L will permit before the rope will have to slide through the clamps $H\,H'$ of the governor. Now as the distance through which the spring can be compressed, plus the movement of the end of G', is only a few inches, it follows that unless the car is stopped very short, the rope L must break if it cannot slide through the clamps $H\,H'$. The distance in which the car will stop is always considerably more than the compression of the spring plus the movement of the end of G'; hence, while it is necessary for $H\,H'$ to clamp the rope tight enough to move G', the pressure must not be so great as to prevent the rope from slipping. For the same reason, in order to make the safety governor reliable it is necessary that the operating rope shall be in just as good condition as the elevator lifting ropes. The failure to inspect this rope properly and make sure that it is at all times in perfect condition has been a prolific cause of accidents.

The jaws of the safety plank and the wedge N should be kept clean and in proper adjustment at all times. As the guides M have to be kept well lubricated, it can be easily seen that if the safety jaws are neglected they will soon become clogged with a mixture of grease and dust, and this may give a considerable trouble by causing the wedge to stick to the side of the guide and thus go into action when everything else is running in

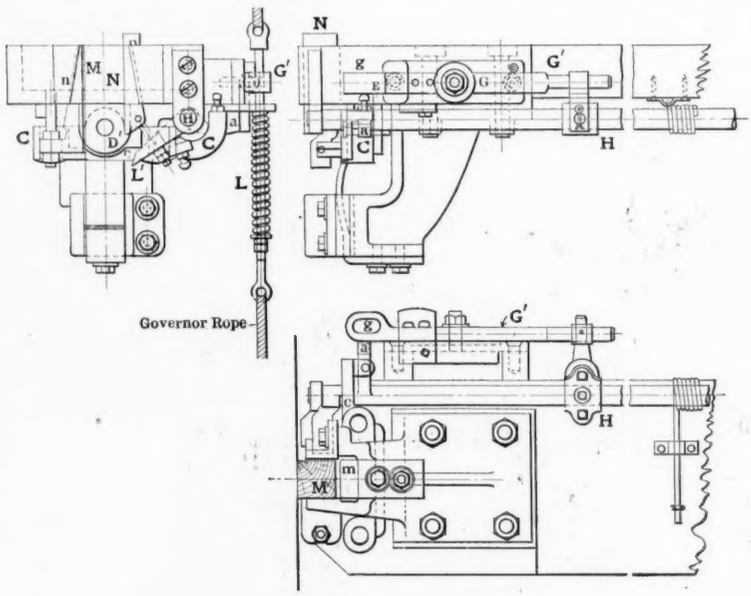


FIG. 27

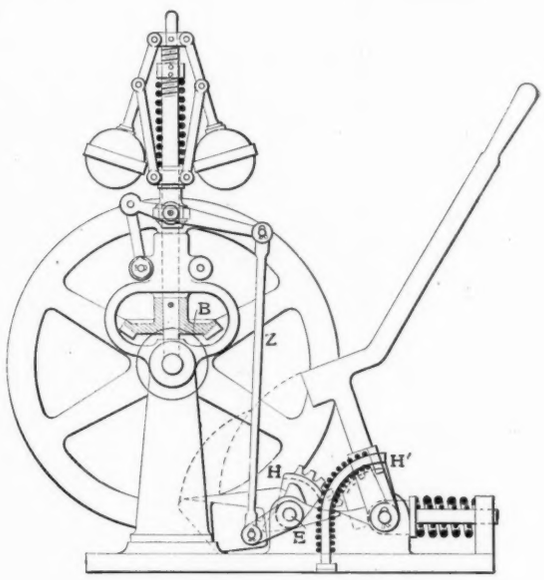


FIG. 28

proper condition. The wedge *N* and the adjusting wedge on the opposite side of the guide will wear away gradually; therefore, the latter must be set up as often as required to keep the clearance between the guide and the safety jaw of the proper amount. If the clearance is too great the wedge *N* is liable to not catch firmly when called into action, and if the clearance is too small, the safety is liable to act without cause.

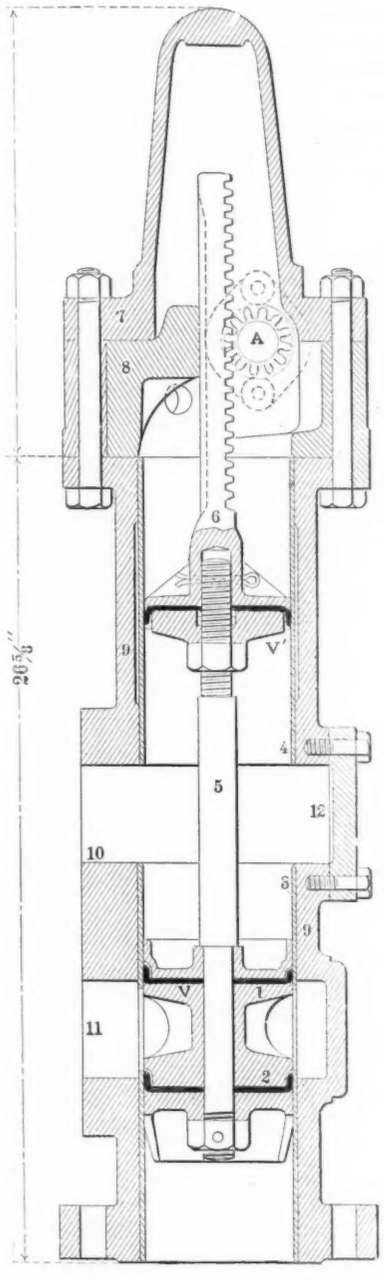


FIG. 29

The operating valve shown in Fig. 25 is the same in general principle as the one shown diagrammatically in Fig. 7, but has several details of construction that are not illustrated in the latter. Its actual design can be readily understood from Fig. 29 which is a sectional elevation of the valve and the casing. The casing is made in three parts, marked *7*, *8*, and *9*. The first forms the top, and provides a dome into which the rack *6* on the end of the valve rod can rise as the valve is lifted by the rotation of the pinion on the end of the shaft *A*. This shaft carries at its outer end the hand-rope sheave shown at the side of the valve in Fig. 25. The parts *7* and *8* are divided at the center of the shaft *A* and form a bearing for the latter. The lower part *9*, which is the valve casing proper, has ports *10* and *11* for connection with the lower end of the circulating pipe and the lower end of the cylinder in the manner indicated by Fig. 7. The part into which the circulating pipe is connected forms a separate casting in Fig. 25 and the casing *9* is bolted to it.

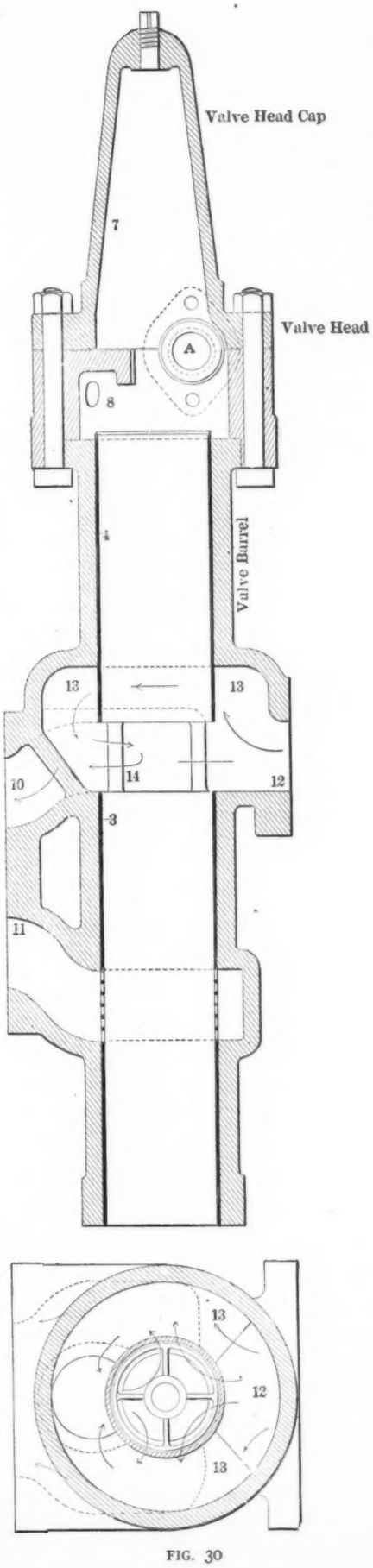


FIG. 30

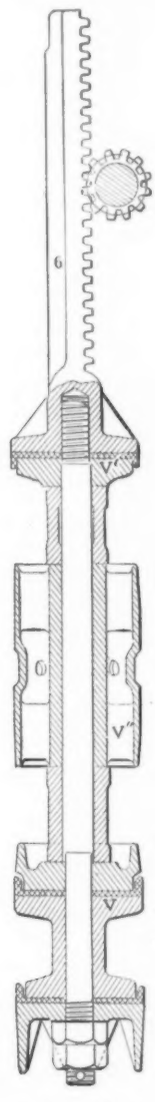


FIG. 31

The port *12* in part *9* of the valve casing is for the purpose of connecting with the pressure-water supply if for any reason it is not desired to have this connection made in the circulating pipe. The valve casing is lined with brass tubing *4* and *3*. The former is simply for the purpose of providing a smooth surface for the cup packing of *V'* to slide against, but the latter is provided for the additional purpose of making ports of such a character that the cup packings of *V* may be able to slide

over them readily. If the ports were large openings, the packings could not pass over them because on the up movement they would be caught by the edges of the ports. With the brass linings this trouble is overcome by perforating the brass with a large number of small holes, about one-quarter of an inch in diameter. The combined area of the holes is much larger than would be required in a single port, this increase in opening being provided so as to reduce the friction of the water running through the holes by reducing the velocity of flow.

The pressure of the water tends to force the valve piston V' upward, and the other piston V downward; both pistons being

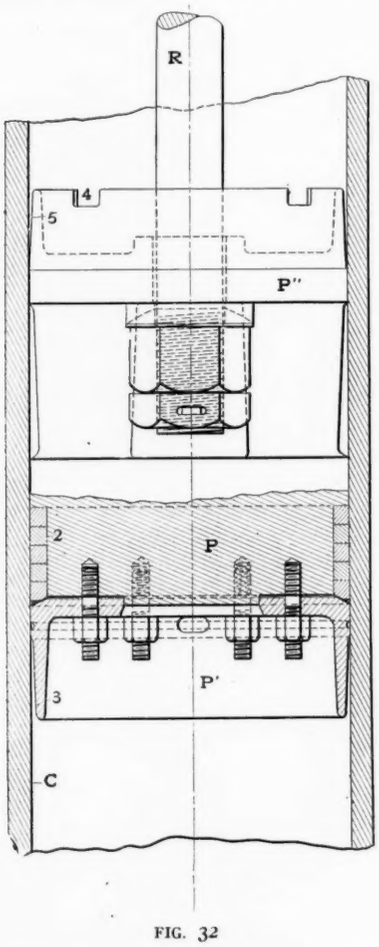


FIG. 32

of the same diameter, the valve is balanced. The force necessary to move the valve is considerable, however, as the friction of the cup packings is great, being produced by the pressure of the water acting upon the entire surface of the leather in contact with the brass linings of the valve casing. On this account the pinion on the shaft *A*, through which the valve is moved, is made very small, while the hand-rope sheave is large—about 20 inches in diameter—so that while the valve travels a few inches in either direction the hand rope has to be pulled through a distance of from two to four feet, according to the size of the valve and the speed of car. For high car speeds the hand rope movement is increased, so that the automatic top and bottom stops may be able to arrest the movement of the car without making the stop abruptly.

In looking at the lower end of valve piston V it will be noticed that the lower head that clamps the packing *2* is made tapering; this is done so that in moving the valve down, to stop the car on the up trip, the outlet from the lower end of the cylinder may not be closed so quickly as to produce a violent stop. Even with this precaution it is possible for the operator to close the valve too rapidly; hence, in addition to this tapering of the main valve, a check valve is inserted in the passage that connects the valve casing with the cylinder. This check is directly under the lower end of the circulating pipe, so that if the operator closes the valve too suddenly the descent of the piston in the cylinder will not be arrested instantly, but will continue its movement, and force the water under it to pass through the relief check valve into the circulating pipe and thus to the upper end of the cylinder.

If the operator moves the hand rope so quickly on the down trip as to produce a violent stop, the piston will continue to rise in the cylinder and the water above it, which cannot pass to the lower end of the cylinder on account of the valve being closed, will be forced back through the inlet pipe *I* to the pressure tank. In this case, as no water can pass into the lower end of the cylinder, the continued upward movement of the piston causes it to leave the water and thus form a vacuum. This vacuum, combined with the pressure in the tank, soon arrests the movement of the car—in fact, in a very few inches—but the stop is not so sudden as to jolt the passengers, as would be the case if there were no relief for the water imprisoned in the cylinder.

One objection to having the connection between the cylinder and the pressure tank through the inlet pipe *I* is that if for any reason the pressure in the tank should drop to zero, as by the springing of a bad leak, the water in the upper end of the cylinder can immediately run out, and with such freedom that if the car were at the top of the building, as in Fig. 25, it would attain a dangerous speed by the time it reached the bottom. This danger can be entirely obviated, however, by placing the pressure tank on the roof, so that the water in the cylinder has to run out against a head, due to the elevation of the tank. To this head is added the pressure of the atmosphere, because as the valve is closed no water can pass into the lower end of the cylinder, and as the piston runs up a vacuum is formed under it, and these combined pressures are sufficient to prevent the car from attaining a dangerously high speed in its descent.

When the pressure tank is placed in the basement the danger above referred to is avoided by using a valve of the type shown in Figs. 30 and 31, the first drawing showing the casing and the second the valve. The difference between this valve and that of Fig. 29 is that it is provided with an additional piston V'', which is called the throttle valve. When this valve is used, the inlet pipe from the pressure tank is attached to the port *12*. When the elevator is stopped, the throttle valve V'' is directly opposite the port *12*, and thus obstructs the flow of water from the port 10. It will be seen that a groove is turned in V'' at the center line; in addition the valve is not made a perfect fit in the valve casing and the clearance afforded by these two features is sufficient to permit water to pass by in as large an amount as may be necessary to prevent too sudden a stoppage of the car, if the operator should

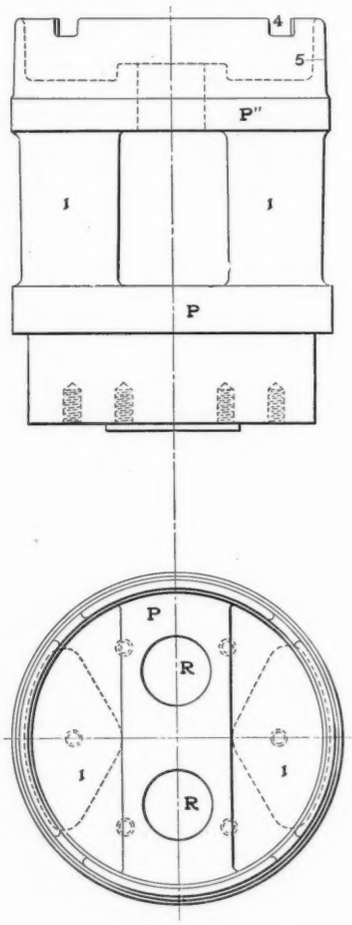


FIG. 33

close the valve too quickly; but it will not allow the water to flow through fast enough to enable the car to descend at a high velocity if the pressure in the tank should fail.

When the valve is moved in either direction to set the car in motion the water passes from the port *12* to the port *10*, through the side ports *14*. Part of the water passes directly from *12* to *14*, and part passes around the upper lining *1* through circular passages *13*, and thence down into *14*, as indicated by the arrows. In this way sufficient opening around the throttle valve is afforded even when the port of the operating valve piston V is only slightly open. The passages *13* and the connection between the

ports *14* and *10* are not easily made out from Fig. 30, but the arrows indicate the course of the water, and these make the construction more easily understood. Fig. 32, which is a section through the passages *13*, taken at right angles to Fig. 30, will serve to illustrate more fully the construction.

The pistons used in vertical hydraulic elevators are made in several designs, some being arranged so as to be packed from the upper end, and others so as to be packed from the lower end. Fig. 32 shows one of the latest designs of pistons arranged to be packed from the lower end of the cylinder, which appears to be the favorite type now. The drawing shows a section through the complete piston, with packing in place, also a section of the cylinder *C*. Ordinary square packing is used, and this is held in position by a follower secured by six bolts. Fig. 33 shows the body of the piston only. The parts *P* and *P''* are made to fit the cylinder, but the intervening section is cut away on opposite sides, so as to afford space for the ends of the piston-rods and their fastening nuts. The top and bottom parts of the piston are connected by the pillars *1* and *1*.

In packing these pistons it is necessary to be careful not to press the packing in too tight, as there is danger of bursting the cylinder by so doing, and even if this much damage is not done the friction caused by the excessive pressure may be so great as to prevent the car from attaining its full velocity. If a hard packing is used, and this is forced into place dry and very tight, the chances are that when it becomes well soaked it will expand enough to burst the cylinder. Bursting hydraulic-elevator cylinders is not a very rare occurrence, and when it does occur it is due to too great pressure of the piston packing against the sides of the cylinder.

The 60,000-volt Kern River Transmission Plant

The Edison Electric Company, of Los Angeles, has practically finished the 60,000-volt transmission line from its 25,000-horse-power hydraulic-electric station of the Kern river, a distance of 128 miles from Los Angeles, consisting of six cables carried on steel towers from 30 to 75 feet high. The power house is equipped with five Allis-Chalmers three-phase generators, each of 3200 kilowatts, direct connected to hydraulic turbines. The canal is about twelve miles long, with a carrying capacity of 700 cubic feet per second and a head of 270 feet. Five sets of step-up transformers are used, each 2250 kilowatts. The transformer station in Los Angeles has four compartments, each containing three transformers of 750 kilowatts each. The power received there is used in the operation of street-railway cars over the Los Angeles city and interurban lines.

An Electrical Indicator for Water Tanks

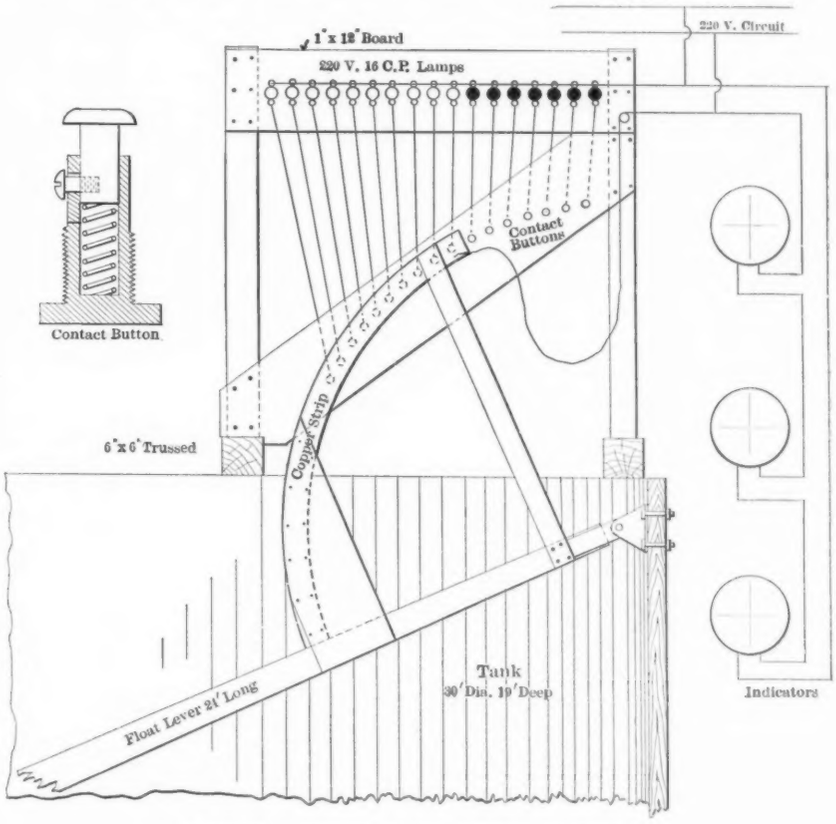
BY C. H. GLASSER

The water supply of the Tennessee Copper Company's smelting plant is delivered to the smelter-service tanks by a belt-driven centrifugal pump operating under a head of 120 feet, and located about 2000 feet from the smelting plant. It was not feasible to regulate the speed of the pump to the variable requirements of the smelter consumption, and it is important that the pump operator, chief engineer and furnace foreman be at all times correctly informed as to the depth of water in the service tanks. This is now accomplished by means of indicators placed at different points in the works, which give by direct readings the depth of water to the nearest even foot.

When such indicators were first contemplated considerable correspondence was conducted with different parties claiming to supply electrical equipment which would accurately give the required indications, but it was found that the apparatus proposed was of such complexity or required such multiplicity of wiring that they were not deemed feasible for the purpose. The depth of the tanks is 19 feet, and we required the apparatus to indicate the approximate hight of water in the tanks at any desired number of stations throughout the works.

One correspondent stated positively that such indications could not be obtained without the use of nineteen separate wires to each station. Another offered a complicated and delicate clock-work mechanism which he claimed would give the necessary indications with the use of two wires. After these proposals had been abandoned the following apparatus was devised and installed by D. J. Kerr, the smelter electrician. It is simple, gives accurate readings at any desired point or number of points, and requires but two wires.

A float lever, the length of which is nearly equal to the diameter of the tank, is attached to the inside of the tank near the top. Close to the fulcrum end of the float lever is attached an arm carrying a segmental copper strip which makes and breaks contact with nineteen stationary copper contact buttons as the float rises or falls. These buttons are attached to a board opposite the moving segmental strip and are spaced the proper distance apart to make successive connections with the copper strip at each foot rise of the float. The contact buttons are connected in parallel through a lamp resistance to one wire of a 220-volt circuit, and the movable copper strip to the opposite wire. These two wires are run to the different points where indications are desired, and ammeters are connected in the line to give the desired readings. The ammeters are calibrated to give full scale deflection with a current of 5 amperes and the scale markings are from 1 to 20 in 0.25-ampere divisions. At the power house a recording ammeter is used, thus giving a chart record of the depth of water at all times.



ELECTRICAL INDICATOR AND METHOD OF WIRING

When the indicator at the pumping station registers a depth of 18 feet, the pump is stopped and remains shut down until the indicator shows 12 feet of water in the tanks, which depth is assumed as a minimum supply.

The greater portion of the water is for blast-furnace use, and it was deemed advisable to erect an indicator near the furnaces, for in case of an accident at the pumping plant it sometimes becomes necessary to cut down the water consumption to the lowest possible point, and at such times the indicator is of especial value to the furnace foreman. The recording indicator at the power house gives a check on the efficiency of the operator at the pumping station.—*The Engineering and Mining Journal.*

Tests of a Producer-gas Engine*

BY G. W. BISSELL, AMES, IOWA

The writer presents briefly in this paper the principal and most interesting results of tests of a producer-gas engine installed for electric-light service in a small town in Iowa.

It was a three-cylinder vertical single-acting engine rated at 150 brake horse-power and guaranteed to develop one-half its power at not to exceed 1.5 pounds anthracite pea coal per brake horse-power; the cylinders were 15x18 inches, and the speed 250 revolutions per minute.

The producer was of the suction type and was furnished with vaporizer, scrubber, etc. The engine was started by compressed air supplied from storage tanks filled by a small gasolene engine which served also to run a small fan-blower for starting the producer and to elevate coal to the storage hopper.

The engine was belted to a 75-kilowatt two-phase Westinghouse generator and exciter. For purposes of brake tests the belt was thrown off and a prony brake applied to a special brake pulley loaned and placed by the makers. All usual and necessary precautions were taken in securing the data.

The results of the brake tests were as follows:

Date.	Run.	Duration, Hours.	Average B. H. P.	Per Cent. of Rating.	Coal as Fired per B. H. P.	Total Efficiency from Coal.
Mar. 1, '06	I	6	40.1	26.8	1.51	15.69
Mar. 2, '06	II	7	82.5	55.0	1.15	17.56
Mar. 3, '06	III	6	156.9	104.2	0.99	19.69

The results of the plant tests were as follows:

Date.	Run.	Duration, Hours.	Total Kw.-H.	Average Kw.	Estimated B. H. P.	Coal per Kw.-H.	Estimated Coal per B. H. P.-H.	Cost of Coal per Kw.-H. at $6.
March 15, 16, '06..	A	14.08	408.6	29.02	48.6	4.10	2.45	$.0123
March 16, 17, '06..	B	12.50	589.3	47.14	74.0	2.23	1.43	.0067

Fig. 1 shows the load curves for the two plant tests. Curve *A* is for the ordinary lighting load. Curve *B* is for the lighting load plus an electrically driven air compressor used for pumping by air-lift and represents normal plant conditions.

Fig. 2 shows the economy and efficiency curve for the brake-load tests and *A* and *B* thereon are from "estimated coal per brake horse-power-hour" in the plant tests.

The heat balance for run III of the brake-load tests is given below.

The American consul at Roubaix has reported on the tendency in France toward the utilization of electricity for industrial purposes, energy being now obtainable at reasonable prices. The success recently achieved by a company at Lille has led to the establishment in Lille or environs of four further plants of large capacity. The latest is a plant of 25,000 horse-power at Marquette.

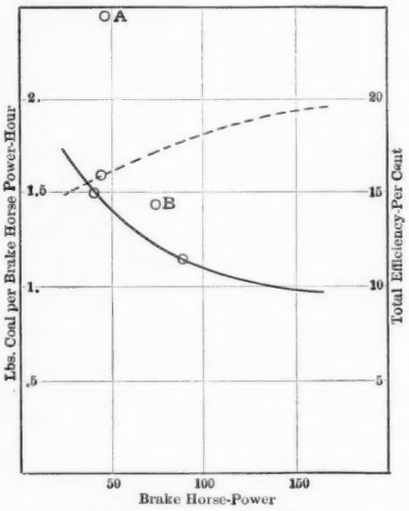


FIG. 2. ECONOMY AND EFFICIENCY CURVE FOR BRAKE-LOAD TESTS

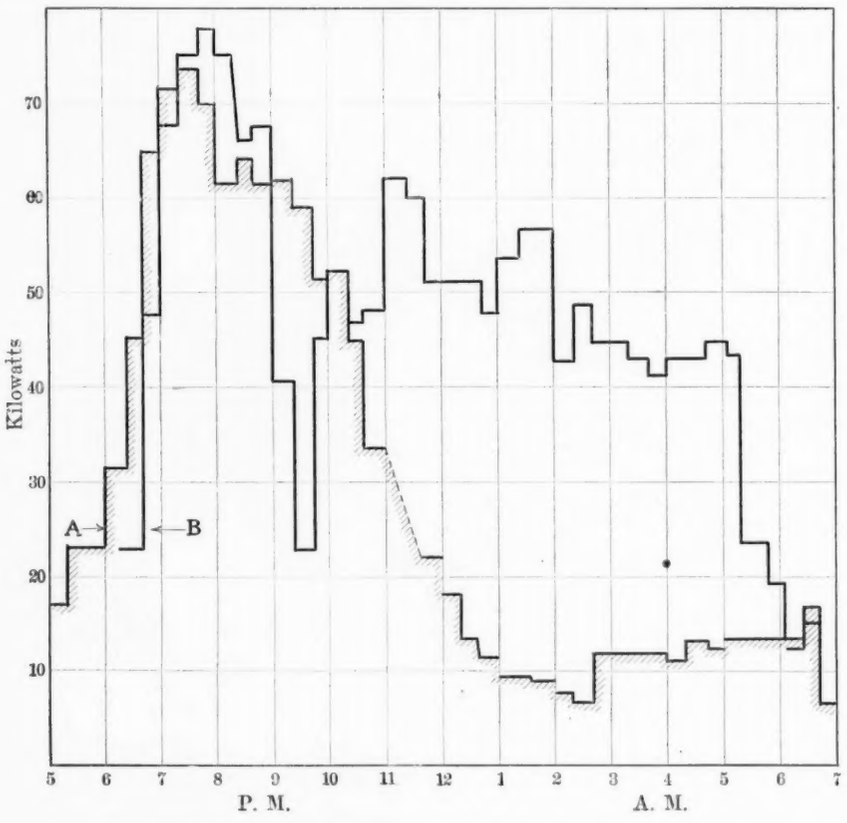


FIG. 1. LOAD CURVES FOR PRODUCER-GAS ENGINE TESTS

HEAT BALANCE AT FULL LOAD.

B.t.u. in coal as fired		100 %	12,143,500
B.t.u. in B. H. P.	2,391,524	19.69	
B.t.u. in unburned coal in ash	1,382,000	11.46	
B.t.u. in radiation from producer	285,000	2.33	
B.t.u. in radiation from scrubber	39,060	0.31	
B.t.u. in vaporizer	79,000	0.65	
B.t.u. in scrubber water	887,000	7.15	
B.t.u. in jacket water	2,539,000	20.9	
B.t.u. in exhaust gases	3,856,000	5.55	
B.t.u. unaccounted for	674,976	5.55	
	12,143,500	100.00	12,143,500

*Read at the fifty-seventh meeting of the American Association for the Advancement of Science, mechanical science and engineering section, held at New York, December 27, 1906, to January 2, 1907.

ELEMENTARY LECTURES ON ELECTRICAL ENGINEERING—XIV

Counter Electromotive Force in a Direct-current Motor Armature—Torque or Angular Pull of the Armature—Relation between Torque, Speed and Power

When a direct-current motor is in operation, the wires around the periphery of its armature "cut" the magnetic lines of force produced by the field magnet exactly as in the case of the dynamo. Consequently, an electromotive force is induced in each wire, as in the dynamo armature. This induced electromotive force is in opposition to the flow of current due to the electromotive force of the supply circuit, and tends, therefore, to keep down that flow of current. Reference to Fig. 41 will make this statement clearer. Here a single loop of wire, connected directly to a source of electricity, is shown on the armature core. With current flowing in the loop in the direction indicated by the arrows *c, c, c, c, c, c, c*, a magnetic field is set up in the direction indicated by the large arrow marked "Direction of Armature Flux." With the field magnet energized so as to produce a field in the direction indicated by the large arrow *F*, the reaction between the two fields will turn the armature core in the direction indicated by the arrow *R*. As the core turns, the upper wire of the loop will cut the flux under the south pole of the field magnet, and the other side of the loop will cut the flux under the north pole; the result will be the induction of an electromotive force in the loop in the direction indicated by the small arrows marked *e*. These, obviously, are in opposition to the flow of current produced by the electromotive force of the supply source. What is true of one loop of the armature winding is true of every other loop, namely,

Any flow of current which reacts on the field magnetism so as to produce rotation of the armature will be opposed by an electromotive force induced by the cutting of the field-magnet flux due to such rotation.

The electromotive force induced in the armature wires by their travel through the field set up by the field magnet is called "counter electromotive" force for the reason that it is counter to the "impressed" or supply e.m.f.

The actual flow of current in the armature of a direct-current motor is that due to the difference between the impressed and the counter electromotive forces. Thus, if a motor is running on a 110-volt circuit, the "impressed" e.m.f. will be 110 volts; now if its counter electromotive force be 105 volts, the net e.m.f. which causes current to flow in the armature will be five volts. If the resistance of the complete armature circuit be 0.1 ohm, the total current in the armature will be $5 \div 0.1 = 50$ amperes. The counter electromotive force of a motor, like the e.m.f. of a dynamo, is proportional to the speed of the armature, the number of armature conductors and the strength of the magnetic field in the air-gaps between the armature and the field-magnet pole-faces. Expressed in the shape of a formula,

$$\frac{\Phi\, w\, s\, p}{100{,}000{,}000\ n} = e, \qquad (1)$$

in which

Φ = Number of magnetic lines of force under one pole-face.
w = Total number of conductors around the periphery of the armature,
s = Revolutions per second,
p = Number of field-magnet poles,
n = Number of paths through the armature winding and
e = Volts of counter e.m.f.

By transposing this formula so as to solve it for revolutions per second, the following is obtained:

$$\frac{100{,}000{,}000\ n\, e}{\Phi\, w\, p} = s. \qquad (2)$$

From the latter formula it is evident that the speed of a motor supplied with current at constant potential varies directly with its counter e.m.f.; also that with other conditions fixed, the stronger the magnetic field, the lower will be the speed of the motor. Conversely, the weaker the field, the higher will be the speed. From this one might argue that with no field at all, the speed would be infinite, but this, of course, is absurd. Weakening the field will increase the speed up to the point where the increase in counter e.m.f. due to the increased speed cuts down the armature current below the value necessary to give the requisite pull at the armature periphery. When this point is reached, any weakening of the field will reduce the speed of the armature.

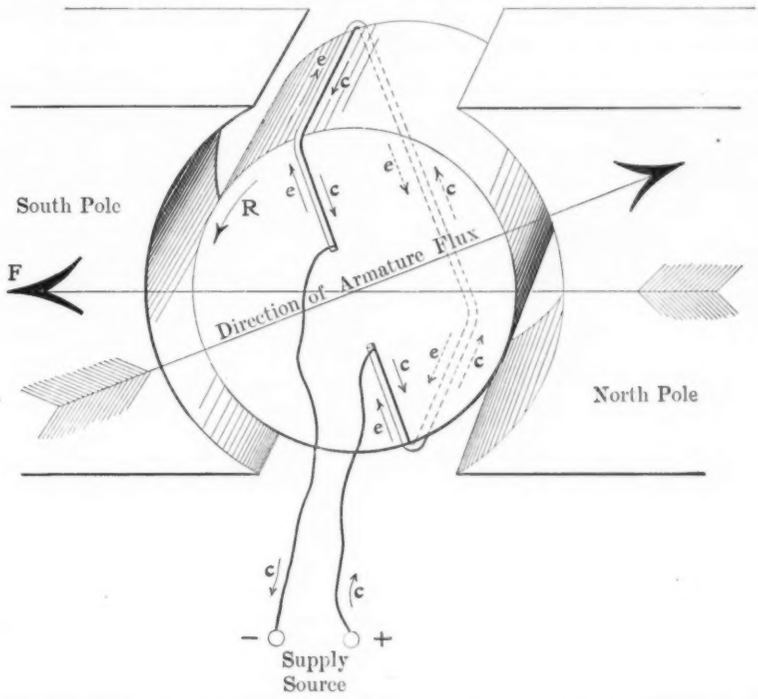


FIG. 41. SHOWING RELATIVE DIRECTIONS OF ARMATURE CURRENT AND COUNTER ELECTROMOTIVE FORCE OF A MOTOR

The "pull" of a motor armature is directly proportional to the strength of the magnetic field, and to the strength of the armature current, the number of armature conductors being fixed. In a field of constant strength, therefore, the pull of the armature depends on the amount of current passing through the winding. This "pull" is expressed as "torque" and measured in pound-feet. A torque of 100 pound-feet, for example, is equivalent to a pull of 100 pounds at a radius of one foot, or 50 pounds at a radius of two feet, or 200 pounds at a radius of six inches, and so on. In other words, the actual pull at the periphery of the armature, in pounds,

multiplied by the radius of the armature in feet or parts of a foot, is the torque in pound-feet.

The torque at the periphery of an armature, or at the distance from the center of the shaft at which the central layer of wires is located, is equal to the current flowing in each wire × number of armature wires × magnetic lines of force under each pole-face × number of field-magnet po'es ÷ 852,200,000.

Expressed in formula shape, the torque in pound-feet is

$$\frac{\Phi\ p\ w\ i}{852{,}200{,}000} = T.$$

The amperes in each conductor are equal to the total armature current divided by the number of paths through the armature winding. Representing the number of paths by n and the total armature current by I, the formula for torque may be written:

$$\frac{\Phi\ p\ w\ I}{852{,}200{,}000 \times n} = T. \qquad (3)$$

When the number of paths through the armature winding is the same as the number of field-magnet poles, which is often the case, the p in the numerator and the n in the denominator of the formula cancel out.

The torque of the armature must be just sufficient to overcome the load imposed on the motor. If the torque is greater than the load requires, the speed of the armature will increase until the increase of its counter electromotive force reduces the current and the increase of speed increases the load to the point of equilibrium between load and torque. If the torque is insufficient for the load, the speed will diminish until equilibrium is established. This is on the supposition that the motor is supplied with current at constant potential. An example will probably make this clearer.

Suppose a four-pole motor to have 480 armature wires, with four paths through the armature winding; a field-magnet flux of 1,000,000 lines per pole; the impressed electromotive force at the terminals to be 120 volts, and the resistance of the armature circuit 0.15 ohm. Also assume that the armature diameter is 12 inches, making the radius one-half foot. As there are four poles and four paths through the armature, these two factors cancel each other and need not be considered. According to the second torque formula (3) the torque per ampere of armature current will be

$$\frac{11{,}735 \times 1{,}000{,}000 \times 480}{10{,}000{,}000{,}000{,}000} = 0.56328$$

pound-foot, and as the radius of the armature is one-half foot, the peripheral pull per ampere of total armature current will be

$$0.56328 \div 0.5 = 1.12656$$

pounds.

Now suppose that the load is such as to require a peripheral pull of 45 pounds. The pull per ampere being 1.12656 pounds, it will require $45 \div 1.12656 = 40$ amperes to give a pull of 45 pounds. The resistance of the armature current being 0.15 ohm, the electromotive force necessary to force 40 amperes through it will be $40 \times 0.15 = 6$ volts. Deducting this from the impressed voltage of 120 leaves 114 volts for the counter e.m.f. of the armature to balance. The speed necessary to generate this counter e.m.f. is readily computed by means of formula (2). The factors n and p cancel out, as already explained; the magnetic flux is 1,000,000; the number of wires 480, and the counter e.m.f. e is 114. Substituting these values in the formula, the result is

$$\frac{100{,}000{,}000 \times 114}{1{,}000{,}000 \times 480} = 23.75$$

revolutions per second, or 1325 revolutions per minute.

Now, while the motor is running at this speed, suppose that the load be changed so as to require a peripheral pull of 180 pounds instead of 45. The pull being four times as great, the requisite armature current will be four times as great, the other conditions being unchanged. The current to get 45 pounds pull was 40 amperes; consequently, to get 180 pounds pull, the current must be 160 amperes, and the electromotive force necessary to force this current through the armature resistance of 0.15 ohm is $160 \times 0.15 = 24$ volts. Deducting this from the impressed e.m.f. of 120 volts, leaves 96 volts to be balanced by counter e.m.f. Applying formula (2) it will be found that the speed at which the armature will run in generating 96 volts is

$$\frac{100{,}000{,}000 \times 96}{1{,}000{,}000 \times 480} = 20$$

revolutions per second, or 1200 revolutions per minute. In other words, increasing the load until the pull at the armature periphery has increased four-fold has caused a reduction in speed from 1325 to 1200 revolutions per minute.

RELATION OF LOAD, TORQUE AND SPEED

The increase of armature torque four-fold, however, in the example just cited, was not caused by a four-fold increase of load as measured in power. If the speed had been kept up by increasing the voltage at the terminals of the motor so as to keep its counter e.m.f. constant in spite of the increase of current through the winding, then the increase of load represented by the four-fold increase in torque would have been also four-fold. In the case used as an example, the load was 6.113 horse-power in the first instance (when the armature current was 40 amperes) and 20.59 horse-power in the second instance, or only a little more than three times as great.

The relation between power, torque and speed is simple when one once has analyzed it. Power is the product of force and velocity; torque is proportional to force; speed is proportional to velocity, and in some instances identical with it. Hence, power is directly proportional to the product of torque and speed. Thus in the previous example, with an armature current of 40 amperes, the speed of the armature was 23.75 revolutions per second and the torque was 22.5 pound-feet, while the horse-power was 6.113; in the second instance the speed was 20 revolutions per second, the torque 90 pound-feet and the horse-power 20.59. Multiplying the speed and torque together and dividing the power by the product will give the same result for both instances, thus:

$$\frac{6.113}{23.75 \times 22.05} = 0.0114,$$

and

$$\frac{20.59}{20 \times 90} = 0.0114.$$

If the reader wishes to check up the figures for horse-power, that is easy. The counter e.m.f. in the first instance was 114 volts and the armature current 40 amperes; the watts of work were, therefore, $114 \times 40 = 4560$. The horse-power was $4560 \div 746 = 6.113$. In the second instance the watts were $96 \times 160 = 15{,}360$ and the horse-power $15{,}360 \div 746 = 20.59$.

The relation between torque, speed and horse-power is given by a very simple formula, as follows:

$$0.011424\ s\ T = H.P. \qquad (4)$$

Its derivation is also simple. The foot-pounds per second done by a revolving armature are equal to the peripheral pull in pounds multiplied by the peripheral velocity in feet per second. The velocity is equal to the revolutions per second multiplied by the circumference in feet, and the latter is equal to the radius in feet multiplied by 6.2832. Consequently, the foot-pounds per second are equal to

$6.2832 \times$ *rev. per second* $\times$ *radius* $\times$ *pull.*

The radius multiplied by the pull equals the torque, so that the latter can be substituted for the two factors composing it, thus:

$6.2832 \times$ *rev. per second* $\times$ *torque* = *foot-pounds per second.*

One horse-power is equivalent to 550 foot-pounds per second; consequently, dividing foot-pounds per second by 550 will give horse-power.

Hence,

$$6.2832 \times s \times T \div 550 = H.P.$$

and if we divide 6.2832 by 550, to reduce the number of fixed values or "constants," the result is 0.011424, giving formula (4). This may be transposed to solve for either torque or speed, as follows:

$$\frac{87.54 \times H.P.}{rev.\ per\ sec.} = torque. \qquad (5)$$

and

$$\frac{87.54 \times H.P.}{torque} = rev.\ per\ sec. \qquad (6)$$

As already explained, $6.2832 \times$ revolutions per second $\times$ torque = foot-pounds per second. Now watts divided by 1.3563 also equals foot-pounds per second. Consequently, watts $\div 1.3563 = 6.2832 \times$ revo-

lutions per second $\times$ torque, and this reduces to

$$Watts = 8.522 \times s \times T.$$

The useful watts in a motor armature are the product of the armature current and the counter e.m.f., the difference between the counter e.m.f and the impressed e.m.f. being used up in the armature losses. Thus,

$$Watts = e \times I.$$

Substituting for e its equivalent as given by formula (1), the result is

$$\frac{\Phi\ w\ s\ p\ I}{100{,}000{,}000\ n} = watts;$$

consequently,

$$\frac{\Phi\ w\ s\ p\ I}{100{,}000{,}000\ n} = 8.522\ s\ T.$$

Canceling s, as it occurs in both of the equivalent expressions, and transferring the constant 8.522 to the denominator of the left-hand part of the equation, the result is

$$\frac{\Phi\ w\ p\ I}{852{,}200{,}000 \times n} = T,$$

and this is identical with formula (3).

EXAMPLES

A few examples will assist materially in obtaining a grasp of the relations just explained.

I. A 110-volt motor has four field-magnet poles, two paths through the armature winding and 400 wires around the armature. The magnetic flux per pole is 1,000,000 lines. When the speed is 750 revolutions per minute, what is the counter electromotive force?

Substituting the specified values in formula (1), we get

$$\frac{1{,}000{,}000 \times 400 \times 12.5 \times 4}{100{,}000{,}000 \times 2} = 100$$

volts.

II. With the motor running at the speed named and therefore generating 100 volts of counter e.m.f., what current will pass through the armature winding if its resistance is 0.2 ohm?

The impressed e.m.f. is 110 volts and the counter e.m.f. 100 volts; there is absorbed in the winding, therefore, 10 volts. The resistance being 0.2 ohm, the current will be $10 \div 0.2 = 50$ amperes.

III. With this current passing through the armature and the conditions mentioned in the first example, what will be the torque?

The flux being 1,000,000, number of poles 4, number of wires 400, current 50 amperes and number of armature paths 2, formula (3) gives

$$\frac{1{,}000{,}000 \times 4 \times 400 \times 50}{852{,}200{,}000 \times 2} = 46.94$$

pound-feet of torque.

IV. With this torque and running at the speed specified in the first example, what is the horse-power delivered at the shaft?

According to formula (4), substituting the known values:

$$0.011424 \times 12.5 \times 46.94 = 6.7$$

horse-power.

V. Check the last result with the useful watts developed in the armature.

The counter e.m.f. being 100 volts and the armature current 50 amperes, the useful watts will be 5000. A horse-power = 746 watts; hence dividing watts by 746 gives horse-power; $5000 \div 746 = 6.7$ horse-power.

VI. Check the value obtained for torque by formula (5).

The horse-power being 6.7 and the speed 12½ revolutions per second, the torque will be

$$\frac{87.54 \times 6.7}{12.5} = 46.93$$

pound-feet, which agrees practically with the torque value obtained in the third example.

VII. If the diameter of the armature were 12 inches, what would be the peripheral pull in pounds?

The torque being 46.9 pound-feet and the armature radius being one-half foot, the pull will be 93.8 pounds.

VIII. With this peripheral pull, how much pull will each armature wire exert?

There are 400 wires, and $93.8 \div 400 = 0.235$ pound. This, however, is the *average* pull per wire. At any given instant, the wires on that part of the armature circumference not covered by the pole-faces do not exert any pull, while those under the pole-faces exert more than the average pull. The maximum pull occurs in the wires that are passing through the strongest part of the magnetic field.

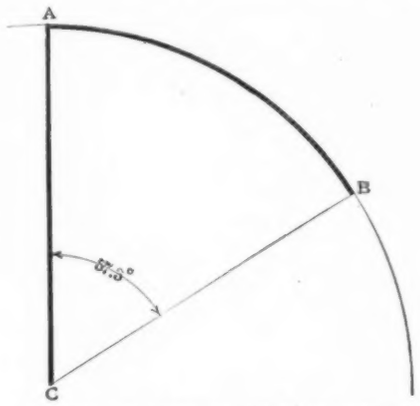
The Velocities of Rotating Bodies

There are several ways of expressing the velocity of a rotating body. The linear velocity or the distance passed through in a unit of time may be given, as when it is said that a crank-pin or the rim of a belt wheel travels at the rate of so many *feet per second* or per minute; the number of complete revolutions which it makes per unit of time may be given, as when it is said that an engine makes seventy-five *revolutions per minute.*

The speed could also be expressed by stating the number of degrees through which the radius of the point in question passed per unit of time, but this would be only 360 times the number of revolutions which it made, for in making revolution it passes through that number of degrees.

The term "angular velocity" is used to express the above quantity but not generally with the single degree as the unit. The unit of angular velocity is the angle through which the radius of a body would move while the point at the end of the radius moved through an arc equal to the length of the radius.

Since the circumference of a circle is 3.1416 times its diameter, the half circumference will be 3.1416 times the radius or half diameter; and since the half circle contains 180 degrees the part of it $A\ B$ in Fig. 1, which is equal to the radius in length, would include an arc of $180 \div 3.1416 = 57.3$ degrees. This unit of angular motion is called a "radian" and the angular velocities expressed in radians, or in the number of times that the body passes through a distance equal to the length of its radius in a given time, is found by dividing the linear velocity by the radius. In this way it is possible to express the angular velocity of a body which does not turn completely around, as that of a connecting-rod, and of a body of any size, the angular velocity being the same for a radius as great as that of the earth or as small as that of an atom, providing the one turns through the same number of degrees per unit of time, or makes the same number of revolutions per second. The angular velocity is used in a variety of problems and must not be confused with the other expressions for the speed of a rotary body. When it is called



A RADIAN, OR UNIT OF ANGULAR VELOCITY

for, divide the linear velocity by the radius:

$$angular\ velocity = \frac{linear\ velocity}{radius},$$

or

$$A = \frac{v}{r};$$

or since the linear velocity is the product of the diameter or twice the radius by 3.1416, or π, and the number of revolutions:

$$A = \frac{2\ r\ \pi\ N}{60\ r} = \frac{2\ \pi\ N}{60} = 0.10472\ N,$$

where N = the number of revolutions per minute,

or $A = 6.2832\ n$,

where n = the number of revolutions per second.

Of twenty-two gas installations in Germany, of a total of thirteen thousand horse-power, working on coke-oven gas, eleven, or 50 per cent., do not find it necessary to purify the gas. The others use bog ore or sawdust. With efficient tar extractors and ammonia washers, tar and dust are readily removed. One of these German installations was using a gas with 0.2 per cent. sulphur without injurious effect on the iron.—*The Mechanical Engineer.*

Centrifugal Pumps

BY HENRY F. SCHMIDT

Although for years the centrifugal pump has been looked upon as a very inefficient method of pumping large volumes of water at low heads, generally not exceeding twenty-five feet, during the past eight or ten years considerable progress has

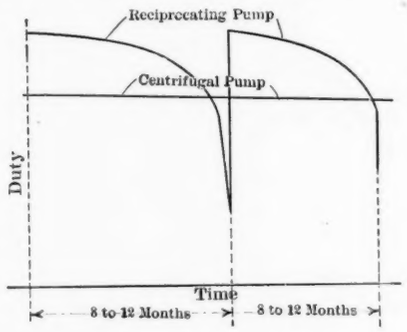


FIG. 1

been made in Europe toward increasing the efficiency of this type of pump, at the same time improving the design so as to deal with very much higher heads. The result is that it is now possible under favorable conditions to obtain efficiencies of from 75 to 80 per cent., even at heads of from five to six hundred feet. This result is accomplished by two expedients, the high pressures being obtained by the use of multiple stages, that is, by delivering the discharge from one pump to the suction of the next, and so on, thus adding together the pressures created in the separate pumps; the high efficiency being in a great measure due to the addition of diffusion vanes placed around the impeller in such manner that part of the kinetic energy imparted to the water by the impeller is converted into the potential or pressure form.

While this work has been going on in Europe for a long period, America has been backward in realizing the possibilities in the development of these pumps, and it is only within the past two or three years that it has received serious atten-

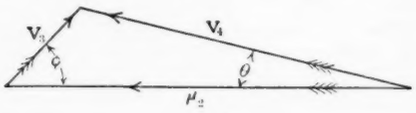


FIG. 2

tion for other than low-lift work. Now, however, a number of American manufacturers have placed pumps of this description on the market, albeit with one exception they have followed the designs of the European builders very closely.

In the light of past developments, it is hardly optimistic to predict that within the next few years the turbine centrifugal pump, driven either by a steam turbine or high-speed gas engine, will replace the present type of plunger pump for municipal water works, hydraulic-elevator service, fire pumps, and, in fact, in nearly every service where the volume of water to be handled is large enough so that a high efficiency can be obtained in the centrifugal pump.

Having made this statement, it is proper to submit the reasons why, as follows: First, small cost; second, small space required; third, the efficiency has been found from experience to remain practically constant over years of continuous use; fourth, absence of valves and plungers which need to be packed; fifth, only one moving part and nothing that can get out of order, consequently practically no attendance needed; sixth, almost "fool proof" construction, it being possible to shut the delivery or suction valves completely when pump is running, without injury to the pump or piping, since the pressure can only raise about 10 per cent. under these conditions; seventh, it is practically automatic, because if the pressure on the line decreases slightly, the volume of water is greatly increased, while if the pressure is slightly increased—showing that the demand is not so great—the volume delivered is reduced, and this entirely independent of pressure-controlling devices or attention from the man in charge; eighth, no more oil is required than for an electric motor of the power required to drive the pump; ninth, practi-

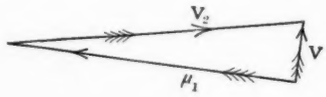


FIG. 3

cally no depreciation need be charged against a centrifugal pump, where the water is fairly clean, other than that required to rebabbitt the bearings when worn; tenth, steady pressure and flow of water.

The result of these conditions is such that in the case of a centrifugal pump to deliver 1600 gallons per minute against a 350-foot head for a hydraulic-elevator plant, notwithstanding that the duty guaranteed for a three-cylinder compound pump was twice that for a centrifugal pump, when the interest on the investment, oil, valve renewals, etc., were taken into account, the centrifugal pump showed itself superior by a little over two hundred dollars a year. This brings out a point to which I wish to call the attention of anyone who may have to compare the yearly costs of a plunger pump and a centrifugal pump. It is the common custom to figure the annual cost for coal on the assumption that the efficiency of the pump remains constant for the entire year, but this assumption is far from true. In the case of the plunger pump, as the valves begin to wear they begin to leak and do not work as well as at first, the leakage increasing quite rapidly toward the end of the life of the valves, until finally one or more valves give way entirely, and there is a sudden drop in the duty of the pump. This is shown graphically in Fig. 1, in which the curved lines represent the duty of a plunger pump, while the horizontal line nearly parallel to the base represents the duty of a centrifugal pump, the efficiency of which remains practically constant. The value of a pump is proportional to the area under this curve, so it is seen from the diagram that the all-round yearly value of the plunger pump is not nearly what it appears to be.

Some will no doubt argue that in a well cared for pump this state of affairs would never exist, but this is extremely doubtful, partly because this wear of valves goes on unnoticed until a valve breaks and causes a pound, and partly because

FIG. 4

the valves are usually rather difficult to get at, and therefore the average engineer will not take the trouble to replace a valve until it becomes absolutely necessary. Furthermore, it must be remembered that in general the man in charge of a plant is only an "average engineer." For this reason I think that in making the yearly estimate of the running costs the plunger pump should be credited with at least 5 to 10 per cent. lower efficiency than is shown on the face of the guarantee.

Owing to the fact that it is very difficult to obtain a cheap and economical steam turbine to run at the speeds required, that is, under 2000 revolutions per minute, little has been done so far in the driving of centrifugal pumps by this type of motor, except those in which gearing is employed. The reason for this is that the makers of pumps have found that the efficiency of the pump is seriously impaired when a high speed is employed, and it is only within the last year or so that

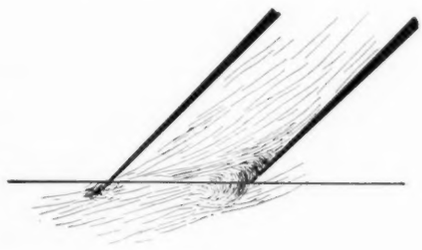


FIG. 5

speeds of even 1800 revolutions per minute have been reached. This drop in the efficiency is found to be caused by an effect similar to what is termed "cavitation" in screw propellers, which refers to the vacuum formed ahead of the screw, due to the water not being able to follow the blades. In pumps, nearly the same thing happens in that the water cannot flow into the impeller fast enough, al-

though in this case it is probably not entirely due to the water not being able to follow the blades, but also due to the blades throwing the water away from them. This is because the water enters the impeller at right angles to the direction of motion, causing shock which has a tendency to pile the water up ahead of the blade and thus throw part of the water away from the blade, instead of causing it to enter.

The velocity diagrams for a centrifugal pump with the blades turned backward

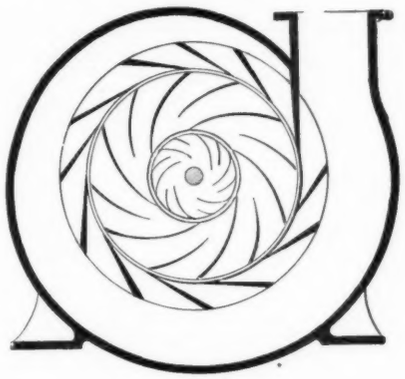


FIG. 6

are shown in Figs. 2 and 3, in which u_1 is the velocity of the inside of the impeller, V is the intake velocity which is at right angles to u_1 and V_2 is the velocity of the water relative to the blade. Fig. 2 is the diagram for the exit, u_2 being the velocity of the circumference of the impeller, V_3 the velocity of the water relative to the frame of the machine. In Fig. 5 is shown the effect which was spoken of above, namely, the piling action due to the water not entering tangentially to the blade.

From this, the writer wishes to make a rather novel suggestion: that of placing guide blades inside of the impeller, as shown diagrammatically in Fig. 6. The object of this will be quite clear from what has already been said. The action of these internal guides is shown in Fig. 4. Here, instead of the water entering the impeller at right angles to the direction of motion, it enters with a greater velocity and at an angle, as shown by V_1. Consequently, the velocity V_2 is made much smaller and the angle between it and u_1 is made greater, so that it is possible by proper design to have the water enter the impeller nearly if not quite tangentially.

The final result of this is that the speed of the inside of the impeller may be increased considerably, thus permitting much higher speed of rotation to be adopted, and still obtain the same efficiency, if not a higher efficiency, for reasons which will presently be given. The velocity which it is possible to give V_1 will probably be from thirty-five to forty feet per second, which means that the blade velocity may be increased a similar amount, although of course it is needless to say that the lifting power of the pump would be reduced, and it would therefore be necessary to place the pump lower, so that the water would nearly run into it by gravity, but this would be objectionable only in comparatively rare cases.

If the water can be made to enter the pump, higher speeds of rotation are advantageous to economy, for, with a given head, the diameter of the impellers may be made smaller, which means smaller rotation losses due to the friction of the water on the disks of the impellers. The power thus lost varies as the cube of the number of revolutions per minute, and the fifth power of the diameter of the impeller, or, very nearly by the formula,

$$HP = \left(\frac{N}{100}\right)^3 \frac{D^5}{1000}$$

in which N is the number of revolutions per minute, and D is the diameter in feet. Consequently, if the number of revolutions is increased and the diameter is decreased, the power lost in rotating the impellers is reduced, and furthermore, which amounts to the same as increased efficiency, the cost is reduced, since the higher the number of revolutions, the smaller the pump, and less interest chargeable against the installation.

Lower cost of construction brings up another suggestion which the writer wishes to make. Instead of casting the impellers and blades in one piece, and finishing the surface by hand, the writer proposes making the blades out of sheet-steel or bronze and bending them to proper shape on a form, then finishing them on an emery wheel, after which they

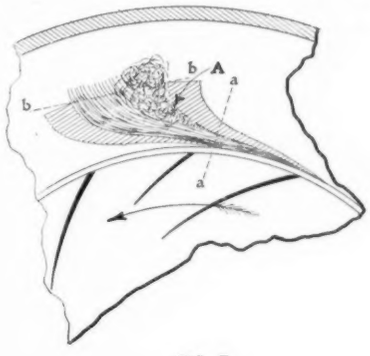


FIG. 7

are set in the mold and cast into the impeller disks and hub when the molten metal is run in the mold. In this way, the cost of finishing the impellers would be materially reduced, which in the end is equal to a gain in efficiency. The diffusion vanes can be made in a similar manner and cast with the casing, and as two sides of the channel formed are polished, the sides would not need to be finished, which of course also applies to the impeller.

Fig. 7 shows the usual form of diffusion vane employed in turbine pumps, and the writer wishes to criticize this form of diffusion vane for not following the natural stream lines of the water as it leaves the impeller. It is evident that the water leaving the impeller would naturally tend to flow in nearly straight lines, and hence in the form of vane shown in Fig. 7, there would be a decided tendency to form eddies at A, which would materially reduce the effective area of the passage at $b\,b$. Now, the area at $a\,a$ should be to the area at $b\,b$ inversely as the velocities at those points, in order that the velocity of the water may be reduced from the high rate at which it leaves the impeller to that allowable in the passages, and thereby the kinetic energy converted into potential energy. If, however, part of the area at $b\,b$ is ineffective, the reduction in velocity will not be so great, or, rather, though the velocity is reduced, the kinetic energy which it represents is only partially converted into potential energy, the remainder being lost in eddies. Of course, there will always be a certain loss from eddies, but the sharp turn at A will tend to increase it greatly.

To avoid this, the writer proposes the form shown in Fig. 8, in which the water can more nearly follow its natural course, and which presents no sharp turns which are bound to form eddies. Another point in this connection is the number of diffusion vanes provided. In most commercial pumps now in the market the number of vanes is from six to twelve, which if applied to an impeller of large diameter makes the passage too wide, and this is also conducive to losses by eddies. If the construction of Fig. 8 were employed, and the blades were formed and cast in the mold, it would be a very simple

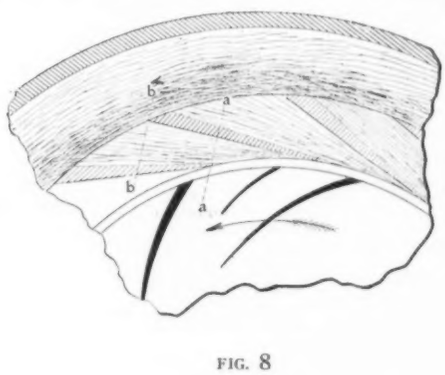


FIG. 8

matter to increase greatly the number of vanes, although of course the opposite extreme must not be approached, since then there would be too great a friction loss.

The reason for the reduced chances for eddies when the passages are narrow is easily seen by approaching the limit when the space between the walls would be so small that a molecule could just pass through them without a chance for side movement.

A producer-gas electric generating station is nearing completion at Johannesburg, Africa, to operate tramways and furnish light and power over an area of 100 square miles.

Boiler Tests with Illinois Coals

Bulletin No. 7 of the University of Illinois Engineering Experiment Station at Urbana (procurable by addressing the University) deals with Fuel Tests of Illinois Coal, made by Professors L. P. Breckenridge, S. W. Parr and H. P. Dirks. The first portion of the bulletin is devoted to explicit and detailed directions for conducting boiler tests, with forms for recording and reporting the observations, descriptions of the apparatus employed, directions for its use, etc., all based upon the code of the American Society of Mechanical Engineers. If it contained nothing else, the bulletin would be well worth sending for by one who is interested in the care, operation and management of steam boilers, but in addition to the above it contains the results of a large number of boiler tests tabulated on the opposite page. In the computation of the results there given the usual correction for quality of steam by proportional weights of steam and water are used. The combustible was computed from the weights of coal and ash and not from the ultimate analysis of the coal, and it is, therefore, in slight error to the extent of the ash which passed over the bridge wall. The basis for the rating of the boilers varied from 10 to 15 square feet of heating surface per horse-power, according to the different types of boilers used. The B. t.u. of the coal, given in the table, were obtained from an analysis of the sample taken during the test.

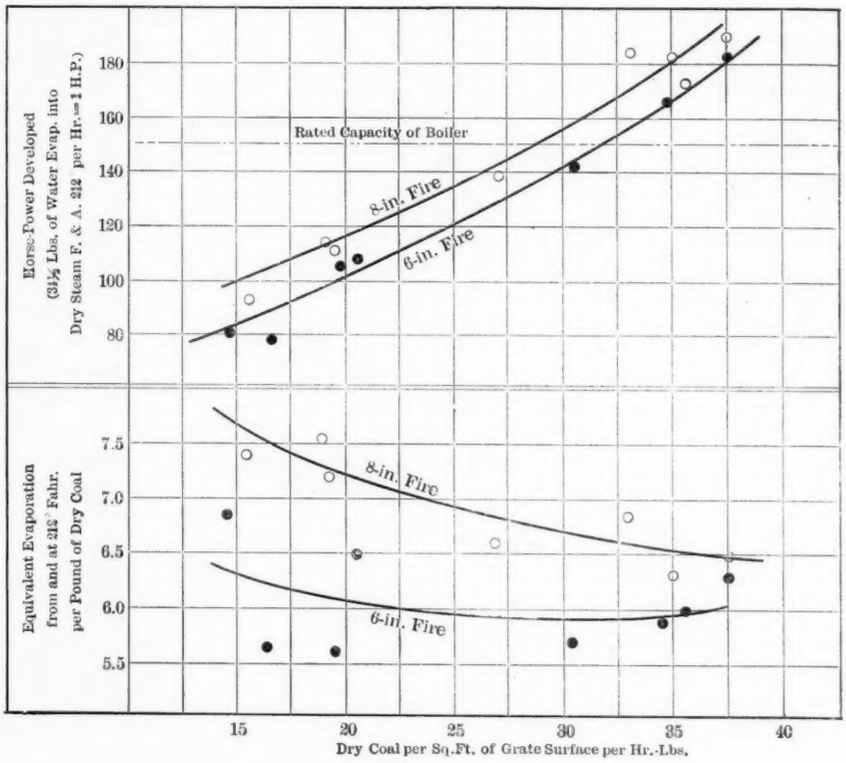
Tests of a small horizontal tubular boiler of 40 horse-power, to determine its performance with varying rates of combustion resulted as follows:

	First Series (5 days). Soot allowed to remain on tubes.	Second Series (5 days). Tubes cleaned each morning.	Third Series (5 days). Soot allowed to remain on tubes.
Equivalent evaporation from and at 212° F. per lb. of dry coal	6.20	7.04	6.23
Dry coal per sq. ft. of grate surface per hour	13.40	9.09	13.40
Horse-power in per cent. of rated capacity	111.00	99.00	115.00
Temperature of escaping gases	627.00	546.00	698.00

Tests to determine the effect of soot deposits on the evaporation of a small horizontal tubular boiler were made on the same boiler as the preceding series and with results as follows:

Dry coal per square foot of grate surface per hour	6.80	9.30	11.00	12.00	14.00
Equivalent evaporation from and at 212° F. per lb. of dry coal	6.20	6.55	6.57	6.37	5.75
Horse-power in per cent. of rated capacity (40)	52.50	87.50	107.50	115.00	122.50
Temperature of escaping gases	432.00	447.00	501.00	516.00	553.00

Tests of a water-tube boiler with chain-grate stoker to determine the relative economy of a 6-inch and an 8-inch fuel bed with various rates of combustion gave results which are best shown by the curves in the accompanying chart. They show that under the conditions of the test the 8-inch fire was the more efficient, giving an equivalent evaporation per pound of dry coal 10 per cent, greater than the 6-inch fire, when operating at the rated capacity of the boiler.



The paper concludes with an excellent article by Professor Parr upon The Chemical Analysis and Heat Values of Illinois Coals, with a description of the process of analysis and of the calorimeters used in determining the heat value.

Asbestos

One of nature's most wonderful products is asbestos, a material, which, in spite of its extensive use, is comparatively unknown to the general public. Prior to 1850, it was looked upon principally as a curiosity, although Charlemagne (Roman Emperor from 800 to 814 A.D.) is said to have had a table cloth made of asbestos, which he cleaned by throwing into fire.

There are two varieties of commercial asbestos, known as amphibole and chrysotile. The former is used only to a comparatively small extent, as the fibers are short and without tensile strength and are, therefore, not suitable for manufacturing many of the asbestos products. Amphibole is used to some extent in cements, but is not well adapted even for that purpose. Chrysotile, on the other hand, has a strong and silky fiber, which adapts it for such materials as asbestos fabrics, household utensils, theater curtains, clothing for firemen, etc.

In Germany, asbestos is known as steinflachs (stone flax) and the miners of Quebec give it quite as expressive a name —pierre coton (cotton stone).

Asbestos is mined in open pits, similar to stone quarries, and although it is found in all parts of the world, the mines in Quebec, Canada, are the most famous, yielding about 85 per cent. of the world's supply of chrysotile. Probably the largest of these mines is that owned by the H. W. Johns-Manville Co. of New York. In 1879 the output of the Quebec mines was 300 tons, which has steadily increased year by year to 50,000 tons in 1905.

The United States Civil Service Commission announces an examination on March 6, 1907, of applicants for the position of steam engineer in the Indian service, the vacancies to be filled being at Fort Peck agency, Montana; Carson, Nevada, and Arapahoe, Oklahoma. The salaries are $720, $800 and $720 per annum, respectively.

Boiler Tests with Illinois Coals made by the University of Illinois, 1894-1905.

Number	Commercial Size of Coal	Number of Tests Averaged	Type of Boiler and Grade	Location of Boiler	Date of Trial	Duration of Trial	Grade Surface	Water Heating Surface	Steam Pressure, Gage	Force of Draft between Damper and Boiler	Temperature of Feed Water	Temperature of escaping Flue Gases	Dry Coal per sq. ft. of Grade Surface per hr.	Horsepower Developed by Boiler	Percentage of Rated Horsepower developed	Equiv. Evaporation from and at 212°Fahr. Per sq. ft of Waterheating Surface per hr.	Equiv. Evaporation from and at 212°Fahr. Per pound of Dry Coal	Equiv. Evaporation from and at 212°Fahr. Per pound of Combustible	B. per pound of Dry Coal	Efficiency of Boiler including Grate
						Hours	Sq. ft.	Sq. ft.	Lbs. Sq. in.	Inches, Water	°Fahr.	°Fahr.	Lbs.	Hp.	%	Lbs.	Lbs.	Lbs.	B t.u.	%
			A. S. M. E. Code No.		1	2	3	7	11	12	20	21	49	65	67	64	70	71	50	73
1	Slack	4†	B. & W. No. 1 & 2, plain grate	Urbana & Cham Elec. Light Plant	June 1894	10.12	52.1	2340	90.2	.220	59.1	498	15.10	108.5	51.7	1.60	4.90	6.38		
2	Pea	1†	do	do	June 1894	10.00	51.0	2264	91.3	.220	60.9	484	13.80	110.8	52.8	1.69	5.44	6.63		
3	Screenings	1†	do	do	June 1894	10.00	51.0	2264	95.9	.200	60.0	464	14.90	114.9	54.7	1.75	5.21	6.18		
4	Lump and slack	1†	do	do	June 1894	10.18	51.0	2264	95.0	.325	60.7	529	20.40	106.4	93.5	2.99	6.50	7.44		
5	Lump	2	Hor. Tub. No. 2, plain grate	Univ. of Illinois, M. E. Lab.	Feb. 1897	8.00	18.7	533	82.1	.405	47.5	549	13.56	52.0	130.0	3.37	7.05	7.78	11430	59.6
6	Pea	4†	National W. T. No. 4 Murphy furnace.	Univ. of Illinois, Cent. Heat Plant.	Mar. 1899	7.25	60.7	2579	108.1	.580	49.1	481	18.43	206.3	82.5	2.76	6.36	7.09		
7	Nut	1*	B. & W. No. 5, plain grate	do	Jan. 1895	10.00	51.0	2450	59.5	.300	49.6	485	18.74	135.4	61.5	1.19	4.89	5.95		
8	Lump	2	Hor. Tub. No. 2, plain grate	Univ. of Illinois, M. E. Lab.	Feb. 1897	8.00	18.7	533	75.8	.470	47.0	567	13.79	52.1	130.2	3.38	6.96	7.53		
9	Lump	3	B. & W. No. 2, chain grate	Univ. of Illinois, Cent. Heat. Plant	May 1901	8.00	28.0	1486	108.4	.600	60.0	571	32.40	135.1	90.1	3.14	5.15	7.17		
10	Lump	5	Hor. Tub. No. 1 plain grate	Univ. of Illinois, M. E. Lab.	Apr. 1894	7.74	18.7	533	70.3	.262	56.3	489	10.38	36.6	91.5	2.37	6.48	7.73		
11	Lump	5*	Hor. Tub. No. 2, plain grate	do	Apr. 1895	10.10	18.7	533	59.0	.200	53.7	680	13.63	45.9	114.7	2.97	6.19	7.44		
12	Pea	3*	do	do	June 1896	9.50	18.7	533	67.8	.163	61.1	508	8.41	24.6	61.5	1.59	5.38	6.25		
13	Pea	1	Hor. Tub. No. 1, plain grate	do	Oct. 1896	8.00	16.6	547	73.0	.200	59.6	494	11.99	35.5	88.7	2.24	6.13	7.32	11070	53.5
14	Pea	3	Hor. Tub. No. 3, 4, plain grate	Urbana & Cham., Water Works	Jan. 1897	8.00	45.0	2260	70.8	.394	128.0	359	10.31	84.6	47.0	1.29	6.29	7.49		
15	Lump	2*	Hor. Tub. No. 4, rocking grate	Univ. of Illinois, Cent. Heat. Plant	Mar. 1895	9.38	30.0	1010	69.4	.219	52.2	560	22.49	135.0	135.0	4.61	6.91	7.87		
16	Lump	5	B. & W. No. 5, plain grate	do	Jan. 1895	8.71	51.0	2450	67.8	.330	50.9	478	20.20	176.4	80.2	2.48	5.91	7.00		
17	Pea	1	do	do	Mar. 1899	8.00	51.0	2450	103.7	.600	49.7	518	27.10	555.7	126.3	3.91	6.92	8.82		
18	Lump	7	Stirling No. 3, plain grate	do	Nov. 1896	7.85	13.5	525	82.0	.250	21.8	544	19.91	51.2	85.3	3.37	6.57	7.63	11633	54.5
19	Lump	1†	B. & W. No. 2, plain grate	Urbana & Cham. Elec. L. & P. Co.	June 1894	10.00	51.0	2264	98.5	.210	59.5	482	15.20	129.1	61.5	1.97	5.74	6.51		
20	Slack	1	do	do	Dec. 1896	8.00	51.0	2264	106.4	.730	45.0	462	23.51	205.8	98.0	3.14	5.92	6.83		
21	Duff	1	Stirling No. 7 & 8, plain grate	do	Mar. 1905	23.60	50.9	2587	110.6	.650	178.5	591	20.78	382.6	73.6	2.55	6.24	8.11	11185	53.8
22	Pea	2†	Nat. W. T. No. 4, Murphy furnace.	do	Jan. 1899	8.00	60.7	2579	84.7	.515	53.0	502	21.04	216.9	86.8	2.90	5.86	7.13		
23	Lump	10*	Hor. Tub. No. 2, plain grate	Univ. of Illinois, Cent. Heat Plant.	Feb. 1895	10.76	18.7	533	65.6	.243	52.2	578	11.34	40.7	101.7	2.64	6.64	8.01		
24	Lump	1*	B. & W. No. 5, plain grate	Univ. of Illinois, M. E. Lab.	Dec. 1894	6.00	51.0	2450	53.6	.300	53.6	.....	24.76	194.2	88.3	2.73	5.30	6.26		
25	Lump	2	B. & W. No. 1, plain grate	Univ. of Illinois, Cent. Heat. Plant	May 1902	8.00	35.0	1486	106.3	.500	70.5	.....	22.70	116.3	77.5	2.70	5.06	6.64	12025	4.06
26	Lump	2	Hor. Tub. No. 2, plain grate	do	Mar. 1897	7.83	18.7	533	77.5	.190	50.6	523	11.63	41.0	102.5	2.66	6.49	7.09		
27	Lump	4*	Hor. Tub. No. 1, plain grate	Univ. of Illinois, M. E. Lab.	Oct. 1895	8.00	16.6	547	63.1	.236	63.6	528	10.66	28.6	71.5	1.80	5.54	6.49	11250	47.6
28	Lump	2	Hor. Tub. No. 2, plain grate	do	June 1896	9.25	18.7	533	69.0	.180	61.6	503	7.37	24.6	61.5	1.59	6.15	6.85		
29	Slack	1	Hor. Tub., plain grate	do	Apr. 1898	8.00	22.5	871	93.0	.320	177.7	535	15.96	73.9	123.2	2.93	7.10	8.60		
30	Lump	1*	Hor. Tub. No. 4, rocking grate	Twin City Ice & Cold Storage Co.	Feb. 1895	9.00	30.0	1010	65.6	.257	48.4	519	18.24	117.0	117.0	4.00	7.38	8.30	11250	63.4
31	Lump	2*	B. & W. No. 5, plain grate	Univ. of Illinois, Cent. Heat. Plant	Feb. 1895	9.50	51.0	2450	68.3	.303	49.8	515	19.59	190.3	86.5	2.68	6.57	7.52		
32	Pea	2	B. & W. No. 2, plain grate	Univ. of Illinois Cent. Heat. Plant	Mar. 1901	12.00	51.0	2264	108.5	.700	69.2	420	16.06	118.5	56.4	1.81	5.00	5.87		
33	Pea	2	Hor. Tub. rocking grate	Urbana & Cham. Elec. L. & P. Co.	Apr. 1898	7.50	25.0	.....	88.8	.300	100.1	403	24.31	118.7	.....	.....	6.74	7.75		
34	Pea	1	B. & W. No. 5 & 6, plain grate	Decatur, Illinois, Water Works	Mar. 1899	8.00	51.0	2450	105.8	.600	53.4	539	29.00	543.7	123.6	3.83	6.34	7.64		
35	Pea	3	Nat. W. T. No. 4, Murphy furnace	Univ. of Illinois, Cent. Heat. Plant	Mar. 1899	8.00	60.7	2579	104.3	.540	52.1	503	15.95	152.2	60.9	2.04	5.42	6.21		
36	Lump	3	B. & W. No. 2, chain grate	do	May 1901	8.00	28.0	1486	102.3	.600	60.6	512	28.60	125.9	83.9	2.92	5.42	6.98		
37	Slack	2	do	do	Apr. 1904	8.00	28.0	1486	119.0	.325	78.5	453	22.40	103.5	69.0	2.41	5.69	7.06	11700	
38	Pea	6	Nat. W. T. No. 4, Murphy furnace.	do	Feb. 1899	8.00	60.7	2579	106.5	.550	49.5	495	18.00	179.7	71.9	2.40	5.68	6.33		
39	Pea	3	Nat. W. T. No. 4, Murphy furnace.	Univ. of Illinois, Cent. Heat. Plant	Mar. 1899	8.00	60.7	2579	110.8	.420	48.3	462	19.21	205.2	82.1	2.75	6.07	6.76		
40	Pea	1	B. & W. No. 5 & 6, Roney stoker	do	Jan. 1904	24.00	51.0	2353	119.4	.260	60.6	559	24.30	463.1	105.3	3.39	6.43	8.12	9975	62.2
41	Pea	4	B. & W. No. 2, chain grate	do	Dec. 1903	8.00	28.0	1486	113.7	.700	63.7	530	34.30	165.7	110.4	3.85	5.96	7.39	11152	51.7
42	Pea	3	do	do	Jan. 1904	8.00	28.0	1486	113.7	.670	63.2	535	34.90	185.1	123.4	4.30	6.53	7.95	11152	56.7
43	Pea	4	do	do	Feb. 1904	8.00	28.0	1486	115.0	.630	72.0	435	17.90	89.1	59.4	2.07	6.15	8.26	11152	53.3
44	Pea	4	do	do	Mar. 1904	8.00	28.0	1486	118.7	.580	73.5	441	19.90	114.9	76.6	2.67	7.11	8.93	11152	62.2
45	Pea	5	do	do	Jan 1904	8.00	28.0	1486	114.4	.170	65.2	464	20.50	91.4	60.9	2.12	5.49	6.23	11287	47.0
46	Lump	4	Hor. Tub. No. 2, plain grate	Univ. of Illinois, M. E. Lab.	Feb. 1897	8.51	18.7	533	75.2	.306	48.0	519	13.00	48.3	120.7	3.13	6.84	7.83	11980	55.1
47		2	B. & W. No. 2, chain grate	Univ. of Illinois, Cent. Heat. Plant	Apr. 1904	8.00	28.0	1486	121.6	.145	72.3	492	21.84	114.6	76.4	2.66	6.47	7.44	11930	52.4
48		4	Nat. W. T. No. 4, chain grate	do	Jan. 1905	9.77	59.0	2513	117.8	.360	79.8	635	30.52	335.8	134.3	4.61	6.43	8.21	12162	51.5
49		4	do	do	Apr. 1905	10.02	55.8	2513	122.9	.370	73.6	584	30.73	331.2	132.5	4.55	6.67	8.10	11537	55.9
50	Pea	2‡	B. & W. No. 2, plain grate	do	Apr. 1901	8.19	35.0	1486	104.9	.620	63.5	462	25.70	111.3	74.2	2.58	4.27	5.18		
51	Pea	2‡	B. & W. No. 2, chain grate	do	Apr. 1901	8.00	35.0	1486	102.8	.640	57.4	519	28.20	176.3	117.5	4.09	6.16	7.75		
52	Pea	2	B. & W. No. 6, Roney stoker	do	Nov. 1905	10.00	51.0	2353	128.4	.290	61.5	608	24.37	238.4	101.4	3.44	6.51	7.79	12591	49.9
53	Lump	1*	B. & W. No. 5, plain grate	do	Jan. 1895	10.00	51.0	2450	55.0	.300	51.5	489	20.78	64.4	74.7	2.32	5.35	6.27	13026	39.7
54	Screenings	1	Murray hor. tub No. 1, 2, 3, rocking grate	Urbana Light, Heat & Power Co.	Mar. 1905	47.66	33.0	1456	98.9	.640	130.6	393	11.27	100.1	66.7	1.58	6.19	8.15	12369	48.3
55	Screenings	1	B. & W. No. 2, plain grate	Urbana & Cham. Elec. L. & P. Co.	Dec. 1896	8.00	51.0	2264	108.2	.740	54.0	487	30.67	230.1	109.6	3.51	5.08	5.95	12413	39.5
56	Slack	1	Hor. Tub. No. 2 & 3 plain grate	Urbana & Cham. Water Works	Nov. 1895	10.00	22.5	1060	74.8	.430	140.0	301	13.62	107.9	63.5	1.76	6.07	7.50		
57	Lump	1‡	Stirling No. 7 & 8, plain grate	Urbana & Cham. Elec. L & P. Co.	Mar 1905	24.00	50.9	2587	112.3	.790	182.8	634	25.42	497.6	95.7	3.32	6.63	8.07	12503	51.0
58	Screenings	1	do	do	Apr. 1905	10.80	50.9	2587	110.7	.790	200.0	616	25.09	507.4	97.6	3.38	6.85	8.39	11427	57.9
59	Screenings	1	B. & W. plain grate	do	Apr. 1905	10.33	64.0	2860	102.0	.330	178.0	565	17.91	166.9	64.2	2.01	5.02	6.59	10299	47.1
60	Screenings	2	Heine chain grate	do	May 1905	8.03	72.0	3160	111.9	.710	63.5	592	23.94	382.7	109.3	4.18	7.66	9.64	11067	66.8
61	Pea	2	B. & W. No. 6, Roney stoker	Univ. of Illinois, Cent. Heat. Plant	Nov. 1905	10.00	51.0	2353	133.2	.226	98.3	593	22.40	230.9	98.3	3.39	7.15	8.58	13251	52.1
62	W. Pea & duff	3	do	do	Nov. 1905	10.00	51.0	2353	132.9	.236	61.0	609	19.96	240.2	102.2	3.52	8.14	8.89	13408	58.6
63	New Ky. Pea	3	B. & W. No. 2, chain grate	do	Apr. 1904	9.33	28.0	1486	117.0	.490	65.0	529	32.93	196.0	130.7	4.55	7.34	8.74	12769	55.5
64	do W. Scre'n	1	do	do	Apr. 1904	20.00	28.0	1486	120.1	.540	62.0	534	34.60	210.3	140.2	4.88	7.50	8.89	13156	55.0
65	do W. Scre'n.	1	B. & W. No. 5 & 6, Roney stoker	do	Feb. 1904	12.00	51.0	2353	117.9	.166	57.8	657	26.55	620.7	132.1	4.55	7.75	9.21	12364	60.5

*Steam assumed dry. ‡Coal assumed dry. †Moisture obtained by drying coal above boiler.

CORRESPONDENCE AND DISCUSSION

Correspondence upon Topics of Interest Relating to Power Is Especially Solicited, and Accepted Contributions Are Paid For—Answer Early in Month if Matter Refers to Preceding Issue

Compression in the Otto Cycle

My attention has been called to an evident oversight in your report of the recent meeting of the A. S. M. E., where you make the following editorial comment (page 42, January issue) in connection with my discussion of the Otto and the Diesel cycles:

"[Professor Wood apparently overlooked the fact that it is impossible to compress in the Otto type of engine to as high a pressure as that employed in the Diesel engine.—EDITORS.]"

You will find on page 953 of the January *Proceedings* of the society that I closed my discussion as follows:

"One would not hold, of course, that the Otto engine can practically compress to the amount of the Diesel and then explode the charge at constant volume. I am referring here to the theoretical cycle; only."

I was careful throughout to refer to *cycles*, and not to *engines*. Furthermore, my mathematical discussion of the Diesel or the constant pressure cycle was not intended to be applicable only to the high-compression pressure referred to by the author in his consideration of the Diesel engine; in fact, it was suggested by a study of the Weiderman continuous-combustion engine. ARTHUR J. WOOD.

State College, Penn.

[The statement quoted by Professor Wood from the printed copy of the *Proceedings* was absent from the copy of his discussion which we abstracted. It is quite possible, of course, that the final sheet of his discussion was not in our possession and the omission escaped notice.—EDITORS.]

Advice on Pump Location

Concerning Mr. Ash's request for advice on pump location, on page 117, February POWER, I would say that he may locate his pump at *P* in his illustration, with satisfactory results; but it is not advisable to pipe steam to the pool, as there will be too much loss through condensation.

For operating the pump from a distance, I would suggest rope transmission, this being the cheapest in the end and the most durable. He should locate a check-valve near the pool, so that the water in the suction pipe will not run back to the pool, and to prevent the pipe becoming empty. Thus, it will not require reprimming. GUS A. JANICKE.

Louisville, Ky.

"Should a Corliss Engine Run Under?"

On page 789 of the December issue Hugh T. Boutell replies to the above query, which was propounded by John I. Baker in November POWER. Evidently these gentlemen are not "ice" men, or at least neither of them runs a double-acting ammonia refrigerating machine, which has the cranks set 90 degrees apart, and both engine and compressor on the same side of the shaft. Otherwise the question would have been: Which side should run on the lower guide? or, To which should the motion be imparted, the compressor crosshead or the engine crosshead? It makes slight difference, but sometimes it is noticeable when the weather is hot and the high pressure runs up to 200 plus, say 60 pounds, which makes 260 pounds to pack against.

It is generally understood that an engine which runs "over" is easier to keep in line than one running under. Certainly, in the former case less loss is caused by friction on account of being able to run the top more loosely that way, as it only results in a guide to side motion, which is *nil* if the engine is in perfect line. On the other hand, the bottom shoe is used as the guide in running "under," but if it is not kept just so, it will either heat if too tight or slap and knock if too loose; caused by the crosshead coming to rest on the lower guide at or near the dead centers, due to the action of gravity and being returned by the action of the angle and force of the connecting-rod.

I known of one Corliss engine that had V guides planed out with the end toward the cylinder 3/16 of an inch higher than the other end, measured from the lower guide. A good fit at the crank center gave a 3/16-inch play vertically at the head center, but the bottom guide and cylinder were in line and the engine ran "over" smoothly and gave good satisfaction. I will leave the reader to surmise what would happen if it were reversed.

With an ammonia system the work is the reverse on the compressor side and consequently the pressure on the guides is also reversed. That is, if one were to stand at the compressor cylinder and watch it run "over," it would be seen that the pressure is on the top guide instead of the lower, as on the engine side running in the same direction. This is caused by the force in the compressor cylinder being opposite to that in the steam cylinder. It always looked to me as if the compressor side is the side to be favored, on account of ammonia being harder to hold than steam, more disagreeable to the nostrils of the attendant, and also more expensive.

I submit the above merely as another view of the question under discussion.

Perry, Okla. SUBSCRIBER.

A question which I should like to touch upon, which has been the subject of much discussion in POWER, is that raised by John L. Baker, in the November issue, and answered by A. W. Blanchard and E. S. Hawkins—"Should a Corliss Engine Run Under?"

The reason given by Mr. Blanchard (in the January issue) is not without some weight, but I think that he will agree with me that the greatest reason why an engine should not be run under is on account of the action upon the frame, due to the stress or thrust caused by the angularity of the connecting-rod. When the engine is running over the thrust is always downward, and the frame can be supported from the foundation; but when running under the thrust is upward and tends to lift the frame. This is very noticeable in an engine of the Corliss type of twenty years ago. With the heavy tangye type it does not make much, if any, showing.

A second reason is, as stated by Mr. Blanchard, the difficulty in keeping the top shoe of the crosshead properly lubricated. A third reason is that the crosshead shoes need not be set out as tightly when running over as when running under, as there is no tendency of the crosshead to lift from the bottom guide. It is often the case with an engine running under with a light load, that an unpleasant pound occurs at the end of the stroke, caused by the crosshead dropping to the lower guide and being forced up again, when the crank passes below the center and steam is admitted behind the piston.

I think, however, that the primary reason why one does not like to see an engine run under is because the motion is contrary to the direction of human natural motion; that is, a man will usually turn a crank "over," as we say, and the reason for his doing so is because he can more easily brace himself against the ground, to push the crank over and pull it up, than to pull it over and push it down.

North Adams, Mass. W. J. P.

Effect of Temperature and Pressure Changes on Lap-riveted Joints

Just after an explosion there is a great deal of talk about the weakness of a lap-joint seam. There was no trouble with these seams until steel plates came into use, which would indicate that steel crystallizes much more rapidly than iron. In many cases, after these steel boilers came into use, these cracks would open up sufficiently to permit leaks, which would show there was trouble before the sheet let go. When it occurs in the middle or rear sheet even this would not be discovered, unless very large, in time to prevent the rupture. There must be a cause for the crystallization and rupture.

If we set up one of these boilers and let it stand without use, its strength would remain indefinitely. If it were fired up and the steam pressure raised to its safe limit, fire, feed, pressure and temperatures kept constant, there would never be any trouble, even with a steel boiler with lap joints. The crystallization can come only by bending.

A lap joint is out of a true circle at the joint, which makes the strain a little different from the rest of the shell, and the double metal at this point makes a stiff place so that any bending, or attempts to straighten or form it into a circle like the rest, must come just under the lap. An indication of what an exploded boiler had to undergo is shown by one report of the operations on the morning of the explosion. The steam pressure was raised to 80 pounds, turned on to the factory, which operation reduced it to 50 pounds, and a little later it was raised to 80 pounds again. Pressure, fluctuating like this, will of itself seek out the weak points to attack.

Let us go over a few more troubles which a steam boiler has in addition to frequently changing pressures. Only a few days ago I was in a boiler-room where I had noticed, on my way thither, first, a perfectly clean stack, and then clouds of smoke issuing from it.

The fireman would shovel a good supply of coal in and the liberation of gas would fill the furnace full of flame and smoke, which in a few moments was gone, and there were left large holes in the fire, especially around the edges. This was a vertical tubular boiler; in a square furnace these cavities occur at the corners. Through these holes large quantities of cold air passed with its cooling effect. Apparently there was no damper, or, at least, it was always wide open; the fireman stood in front of two furnaces with shovel in hand, and kept steam somewhere nearly right by the amount of coal—and airholes—in the furnace.

In some cases where 50 or 60 pounds will keep the engine at speed, a fire is made, steam raised to 75 or 100 pounds, and then the fire allowed to get low and the pressure to go down. This occurs many times per day. In many instances, too, the pressure is regulated by the furnace doors, and the boiler gets big drafts of cold air occasionally. Often, the toughest dose the boiler gets is when cleaned.

The man who will allow big holes in the fire, open the furnace doors to check the fire, and subject the boiler to constant severe changes of extreme heat and cold, will, when the time comes to cool down his boiler to wash out, religiously close his damper tight, as well as all the doors (although he may open the flue door), and thus prevent the slightest cooling effect of air in the boiler or brick-work. Then he will draw off hot water, pump in cold water, etc., until the water in the boiler runs cool, and then draw the water all out and subject the empty shell to the hot brick-work. The shell becomes quickly overheated, and as soon as the manhole can be opened, it is quickly cooled off by a spray of water from a hose, after which it is again overheated by the hot brick and again cooled by being filled with water.

I have seen a number of boilers that had received the above treatment, which showed the tremendous strain to which they had been subjected, by leaky tubes that were frequently re-rolled, leaky seams around the front head, patches and new sheets over the furnace; and the wonder was, what could be the reason these boilers did not let go under pressure.

Another trouble is with the way boilers are fed. If the feed is continuous and just the amount the boiler needs, a circulation is established, the temperature will be constant, and no crystallizing strains will result. Where the feed is continuous the floor through the heater will be the hottest possible and this saves cooling effect. Where there is a water tender it sometimes happens that the feed is kept fairly regular, but this is rare. It is the usual practice, where more than one boiler are supplied, that the attendant will have all the water going into one for a time, and then it is turned on to others, the first being shut off, resulting in two sets of strains, one from unequal temperatures and the other from different load from different water level.

One chief engineer asserted that he had well-paid water tenders, and they kept the water levels within one-half inch of each other at all times. When permission was given to measure the level in the different boilers, it was shown that the extremes varied by four inches. Another chief engineer claims that he does not want his water kept level, as it would cause corrosion on the water line.

The worst cases of feeding are where the pump is started up, and the boiler filled up with cool water (as the heater will not keep up the heat when overworked), and then the pump is stopped, and not started again until the water gets low.

Regulating the draft by hand is another method of changing temperatures violently and crystallizing steel plates. Damper regulators that allow the pressure to change ten pounds with a good fireman is another bad thing.

The large shell boilers with lap seams get the worst of the rough treatment to which steam boilers are subjected, but the surprise is that not more of the stronger type do not let go, too.

The butt-strap is easier on the shell at the joint, because it keeps the shell in a circle, and the bending strains from unequal expansion are spread over a larger surface.

So long as we have unequal expansion from widely varying boiler pressures, hot fires and cold drafts, constant or frequent changes in the water feed, and also in the process of cleaning mentioned, just so long will there be danger of mysterious explosions, even with butt-strap seams.

Waterbury, Conn. W. E. CRANE.

"Choice of Feed-water Heaters"

Edw. Kilburn's letter in February POWER, page 120, on "A Get-at-able Coil Heater," emphasizes how vitally important it is to properly select the type of exhaust-steam feed-water heater best adapted to existing conditions. If the water for boiler-feed purposes contains impurities which possess scale-forming properties, such water should be treated through the agency of an open heater. Without casting reflections upon Mr. Kilburn's coil heater *per se*, I am satisfied that it will be readily seen, from his opening sentence, that the open type of heater should be used in his case.

The function of a feed-water heater ought to be more than simply that of a *heater*, as it should combine with this feature that of *purification*, which is equally essential. Such results cannot be as economically attained with a closed heater, due to the difficulty in cleaning, and the lowering of the heating power as the impurities are precipitated. For instance, compare the labor of cleaning all the tubes in Mr. Kilburn's heater with the simple removal of the trays, or pans, in an open heater. In the open type, too, oil is removed from the exhaust by means of improved separators, to the extent that the resulting condensation is rendered quite pure and suitable for boiler-feed purposes. The condensation amounts to approximately one-seventh of all the water used, and in certain instances this would constitute an appreciable item.

In conclusion, allow me to reiterate that more careful consideration should be employed in the choice of a heater, and that the character of the boiler feed-water should be regarded as the foremost criterion in forming a decision.

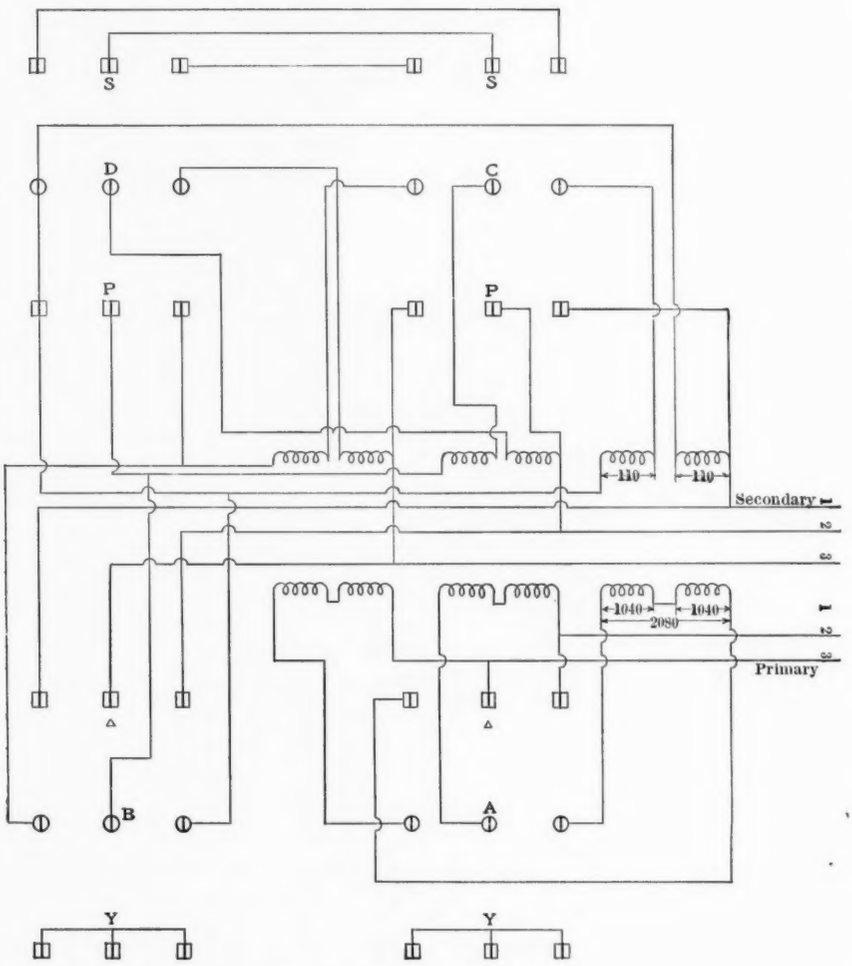
CALVIN B. ROSS.

Mr. Lincoln's Problem in Transformer Connections

The accompanying diagram shows the way in which I solved the problem stated in the February number of POWER. The four switches are indicated by the letters *A, B, C* and *D*, which are inscribed on the central parts of the switches, where the blades are attached. The letter *S* stands for series, *P* for parallel, *Y* for star, and Δ for delta. By placing the switches in the different positions, any one of the combinations can be made. Thus, to connect the primaries in star and the secondaries in delta to deliver 110 volts, put switch *A* at *Y*, switch *B* at Δ, and switches *C* and *D* at *P*.

E. S. LINCOLN.

Boston, Mass.



MR. LINCOLN'S PROBLEM IN TRANSFORMER CONNECTIONS

Low Water and Overheating in Boilers

While looking back through my POWER file recently, my attention was attracted by Mr. Brecknell's article on "The Low Water Emergency," in the June, 1906, issue; also by Mr. Nagle's comments upon the same article in the August issue. I believe that there are projected many opinions upon the subject which are much at variance with each other. Personally I agree with the two gentlemen named, as against E. Cooper Wills in February, 1906, POWER, whose contention is, "That most boilers explode through water being lifted off the crown-sheets with an intensely hot fire, and then allowing a valve to be opened, causing great commotion inside the boiler."

I am not in any way setting up myself as an authority, but I am stating my opinion, and my reason for having it. Like Mr. Nagle I have never yet seen a boiler that has exploded from this cause, and while agreeing that it is bad policy to cause commotion in a boiler, I scarcely believe that if caused by opening valves it would end in the boiler bursting.

We have in daily operation at our works about thirty double-flue Scotch marine boilers, and we work with a water level of 7½ to 9 inches above the combustion-chamber crown-sheets. We employ induced draft, and usually run from 2 to 3 inches water gage, the vacuum varying according to the work the boiler has to do; the steam pressure throughout being 150 pounds per square inch. The boilers have been in use upward of seven years, many of them night and day, and during the entire time there has been no case of burnt sheets or collapsing flues. One pair of boilers are under their work, and work intermittently, feeding an engine which only runs, say, for from thirty to forty-five minutes at a stretch, the engine being required to run this length of time six or seven times a day. During this time it is necessary to stop these engines every five minutes or so; they have to be stopped practically dead, and started up the next minute just as suddenly. Both boilers are piped direct to the engine throttle valve, which is operated by the engine driver, all other valves between the engine and the boiler being thrown wide open.

As before stated, the boilers are under power, and consequently during the intervals of rest the fires are cleaned and got in thorough condition, with the steam going through the safety valve ready for the signal to start. When this is given the engine throttle is opened wide, and the draft pressure goes up to 3½ inches, sometimes 4 inches, and each boiler will burn about 40 pounds of coal per foot of grate, or roughly about 15 hundredweight per hour. This constant stopping and starting, combined with the excessive heat and draft, tries a boiler, and stay-nuts are very often burnt off; but there are no signs of overheating anywhere else.

This treatment is pretty stiff for a boiler with a steady load, and with steam taken as generated, but in this case, with the sudden stoppages, the steam is practically always in a state of commotion, and although on one or two occasions the water has got very low, it has had no result on the boiler. Mr. Nagle speaks of working with a forced-draft pressure of 4½ inches, but these boilers have been worked for experiment under the same conditions of handling with a stoker at each flue, six inches of draft being shown on the water gage, in which case the result was satisfactory, while on power-station work a six-inch draft was used on a battery of boilers for months. The treatment, of course, did the boilers a tremendous amount of harm, but there was no sign of burnt sheets or flues.

Another point which is doubtful is whether an explosion would be caused by the collapse of the flues or sheets unless through great carelessness on the part of the attendant, who should certainly become aware of conditions before they reach a critical stage. Several flues have come under my notice which had dropped from 4 to 4½ inches; they were raised and the boilers put to work again, the last of them eighteen months ago, since when there has been no complaint, so I suppose they are still working satisfactorily. As an object lesson, there are in our plant a couple of flues which are lowered nine or ten inches, yet the boiler from which these were taken was brought down quite safely. There are several other cases at hand

where the crowns of flues have had to be raised, and, in no case has there been an explosion from this cause.

As to the cause of these failures, I can positively state that in six cases it was due to oil being mixed with feed-water, and if oil is carefully kept from the boilers, and they are cleaned so that there is no trouble with priming or foaming, there is not much likelihood of much trouble arising from the boilers, providing they have careful attention and cold air is kept out. With respect to the question of cold water being fed into a boiler near a point where it is overheated, I may point out that the National Boiler Insurance Company, of Manchester, probably the largest company of its kind in England, is having feed-water put in Lancashire boilers at a point about three inches above the furnace crowns, and taking out the old feed-pipes which delivered several inches below this. Evidently from this we may take it that they consider that the danger of a boiler siphoning back through a defective non-return valve and bringing the water level below the crown of the flues is greater than the danger of feeding cold water practically on the top of a flue that may become overheated through low water.

CLYDEBANK.

Sheffield, England.

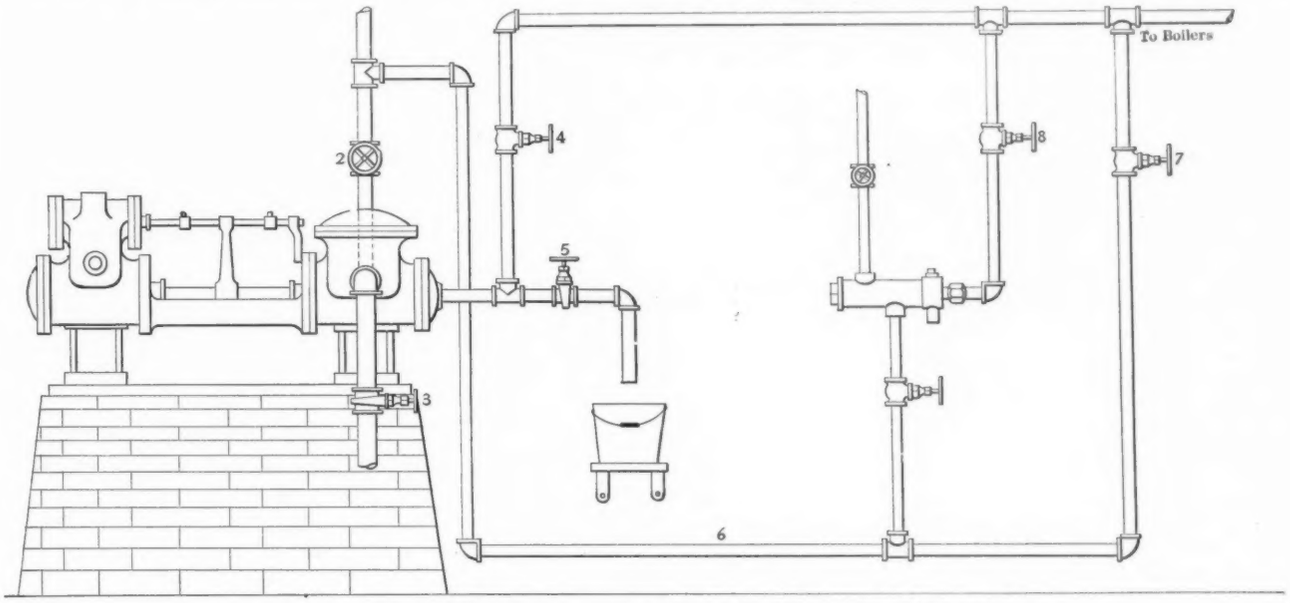


FIG. 1. THE ARRANGEMENT TO WHICH MR. WAKEMAN FINDS OBJECTIONS

"Piping a Steam Pump and an Injector"

I read with interest the article by Mr. Wakeman, in January POWER, pages 30-31, anent the objectionable arrangement of injector and pump pipes and valves. Now, let us take a common-sense view of the changes that he suggests and the anticipated improvements. As to the quick opening of valves, it means a difference of from one to five seconds, only, between the old way and the plan proposed. In most cases like that under discussion, this would not materially matter.

I differ from Mr. Wakeman on another point, also, in that I claim that there is far more friction in most angle-valves than in an ordinary elbow; for the spindle in the valve raises the disk a short distance from the seat and the incoming fluid impinges upon it before it turns, thus retarding the flow nearly as much as in a globe valve.

Should the cross-valves 7 and 8, shown in Fig. 2 in Mr. Wakeman's article (reproduced herewith), become unfit for use,

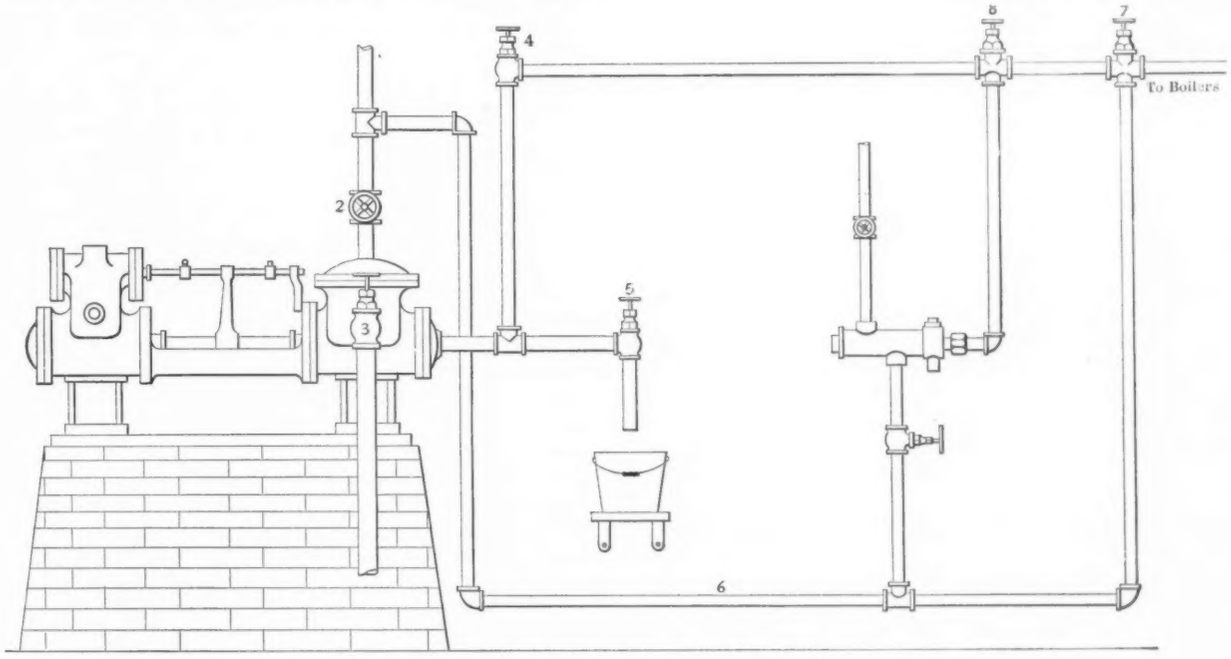


FIG. 2. THE SAME WITH ANGLE- AND CROSS-VALVES SUBSTITUTED

it would be much more expensive and require more work to replace such than those shown in his Fig. 1 (also reproduced). I have proved to my own satisfaction, at least, that for most practical purposes the difference in the quantity of water passing through an angle-valve and a globe valve of the same size, in the same time, is so slight as to be hardly worth considering.

I experimented with two 1-inch and two 2-inch angle- and globe valves, allowing water to flow through both for the same length of time, and the result was practically the same in both instances. A two-inch water pipe runs to a tank in the building where I am employed; I emptied the tank, dried it out, then allowed the water to flow in through a globe valve until the water reached a predetermined hight. I repeated the operation with an angle-valve of the same size, and then with both sizes of the valves. At most there was not more than a couple of seconds in favor of the angle-valves.

If it is advisable to consider such very small percentages of friction, there is certainly less in a good straightway valve than in a cross-valve, for in the latter in most cases, the disk is right in the middle of the valve, where the whole stream of water strikes it with full force. SCOTTY.

A Problem in Transformer-coil Connections

It is desired to supply two lighting systems from a single transformer, containing two 110-volt secondary coils. One of the circuits requires current at 110 volts, while the other calls for a three-wire system delivering 220 volts at the mains with a neutral wire. Either circuit will fully load the transformer, but is not to be supplied with energy at the same time.

The problem is to use one commercial type switch to make the connections between the lighting circuit and the transformer coils. Care must be taken that the right instantaneous polarity exist between these coils when connected to the lighting systems. The primary windings may be ignored. E. S. LINCOLN.

Heating by Gas-engine Exhaust

My firm is adding to the power equipment a 50-horse-power single-cylinder gas engine to be driven by natural gas. The engine will work on the four-stroke cycle, with hit-and-miss governing, and I would like to utilize the heat of the exhaust gases for heating the blacksmith shop, in which the engine will be located. I understand that exhaust gases have been so utilized. I will be grateful to POWER readers for suggestions.

R. MANLY ORR.

Brantford, Ontario.

A "Rap" at Dishonest Manufacturers—"Flues" and "Tubes"—Motor-car Engines and Windmills for Driving Lighting Outfits

I am glad to see that A. W. Peirce in his letter on "Export Mistakes," on page 60 of the January issue, gives dishonest manufacturers a merited rap. The trouble lies largely in that vexed question of the tariff, in that our manufacturers have been protected until they think they are in a position to treat customers just as convenience and whim may dictate. All of them talk about trying to develop a foreign market, but at the same time chuckle in their sleeves and aver that they care nothing about the whole matter as long as Uncle Sam fixes things so they can sell all they make for more in this country than they could get for the same article if shipped abroad. The punishment for manufacturers who treat foreign customers in the way described by Mr. Peirce should be such an exposure that they would lose the lucrative home trade and be compelled to get down to doing business "on the square" before getting any remunerative custom here. Fortunately eventually such things come to light and even after a long time they are almost sure to be found out; so in the end the act brings along its own punishment.

I was interested in reading the distinction between flues and tubes, made by Geo. Mack on page 61 of the same issue. The fact of the matter is that the distinctive appellations for these two articles are still undetermined, notwithstanding all the discussion of the subject that has gone before. If we call all cylindrical openings through which fire or gases pass by the name of "flue," as suggested by Mr. Mack, what shall we call the fire-tube boiler? It is an undisputed fact that a large cylinder through which the products of combustion pass is properly called a "flue," while "tube" cannot be applied to a large cylinder, no matter whether it is used to convey water or not. As the matter stands today it is equally correct to call the 2½- to 4-inch cylinders that carry the gases of a fire-tube boiler, flues or tubes, as the preference of the writer may dictate. There is, however, some distinction in the use of the two designations for the same object. Thus one would hardly speak of renewing the "flues" of a tubular boiler, but would preferably say "tubes." On the other hand, one could speak of cleaning either the "tubes" or the "flues" and speak correctly, though it would probably be better to use the word flue in this connection.

Then again, in speaking of the "things" themselves without reference to their particular use one would say "tubes" and not "flues." Thus, for instance, we speak of a manufacturer turning out lap-welded "tubes," and while one would probably be understood if he said lap-welded "flues" of a certain specified size, he could not be considered as using the most approved form of the vernacular. To sum up the whole thing, it is a matter both of use and size as to which name shall be chosen. If we go through a boiler shop and see a pile of small cylinders lying on the floor we say, "There is a bunch of good tubes." When we look through these tubes in a boiler, we say, "The flues are clean." When we see on the shop floor a number of large cylinders, say 2 feet in diameter, we say, "There are the flues for Smith's Scotch marine boilers." Custom has prescribed her rules and regulations in these things and only time and persistent effort can change them. It makes no difference in the pay envelope which term a man uses so long as he is understood, and until it does make a difference it will be a hard pull to teach us old dogs new tricks. Perhaps some of us have become gray-haired while saying "flues," "tubes," etc., and we may be willing to go down to the grave saying the same. If the educators can get hold of our children, and can themselves agree on what is the proper thing to say, it might be possible to bring up the young ones in the way the lords of expression think they should go. In the meantime the rest of us will stop talking about the matter and get busy with the things that bring in the bread and butter for the little ones.

E. Ralph Estep describes an interesting proposition on page 63, where a small lighting motor is driven by a motor-car engine. A small unit of this kind has a wide field for usefulness, in my opinion, and I believe a discussion of the applications of different classes of machines for use in lighting outfits such as that described would be of general interest to readers of POWER. Take, for instance, wind power. How would it do to employ a windmill for the prime mover, using it to pump water to an elevation or into a hydraulic accumulator for driving the water wheel of a hydro-electric generating unit? How about the question of mill capacity? Can a mill be built to develop, say, 25 horse-power and if so what size should it be? There is, so I understand, a mill on the Pacific coast having a maximum capacity of 200 indicated horse-power, and if this is the case why do the manufacturers not turn out apparatus of more than from 5 to 7 horse-power capacity (about) at the most? Then, what as to the question of economy in operation expense for repairs and attendance? How will these items compare with the gasolene engine and the *coming* alcohol engine? All these items are of vital interest to the small power user and if the western farmer can irrigate his lands with the mill, why can't the same power be made to light his house? No doubt some

of the many POWER readers have had experience with mills for some classes of work and will be able to send in interesting data.

Your editorial on boiler horse-power is to the point. Why not let the transferring of a certain number of heat units by the boiler in a given time be a horse-power, and let this number of units be the same as those equivalent to the heat units corresponding to the number of foot-pounds of work done by an engine or other prime mover in a given time. Thus we could shape a boiler up to fit its place in the plant—provided, of course, that the consumptive qualities of the engine were not abnormal. E. S. HAWKINS.

Chicago, Ill.

Mr. Malcolm's Problem Solved—Troubles with Crude-oil Fuel—Putting Boilers out of Commission—A Suggestion for "Puzzled"

The answer to Geo. W. Malcolm's problem on page 788 of December POWER is that it will require a 110-volt machine. As there is to be no current in the circuit, the matter of line resistance makes no difference; neither does the point of inserting the second machine.

I wish to say amen to all J. W. Homer has to say on page 790 about the "Troubles of the Water Power Man," for his are the troubles that are turning my hair gray. My chief trouble with the turbine has been in the wood bearings, inside the flume, which cut the shaft.

On the steam end I find the most trouble in the boilers. We use crude oil for fuel and the bottom row of tubes blister and crack just above the point of the flame. I would like to hear from others who use oil as to whether they lose more tubes than with solid fuel.

Just how to lay up the steam end is quite a problem. If one knew just how long it was going to be idle, he might proceed along regular lines, but it may be for three months or three days, one can never tell. It must always be ready to serve on six hours' notice, and one dare not dismantle much of it.

I empty the boilers and take the front man-hole plate off and the plate out of the mud-drum. This gives the boiler a chance to dry out. All piston- and valve-rods get a dose of cylinder-oil and the cylinders an extra shot of oil before closing down.

I think "Puzzled," who complains of leaky pipe joints, on page 792, would do well to look for acid gas in his feed-water. As it is a swampy place, the water probably contains some organic acid, which causes his leaky joints.

Mesa, Arizona. FRANK W. CERNY.

Cylinder Wear in Horizontal Engines—Design for Bearing at Piston End of Connecting-rod

Mr. Edwards, on page 765, December POWER, lightly touched upon the topic of piston and cylinder wear in horizontal engines. He intimates that a two-ring piston wears its cylinder less than single-ring pistons do. He also suggests that pistons are held to the bottom of the cylinder in the same way that an unbalanced slide valve is held to its seat, i.e., both have a steam load to carry. Now, it would be easy to cut down the flanges of a piston as in Fig. 1, but what assurance have we that steam will not get between the rings and exert a downward pressure, as shown by the arrow at *a*? Packing rings should

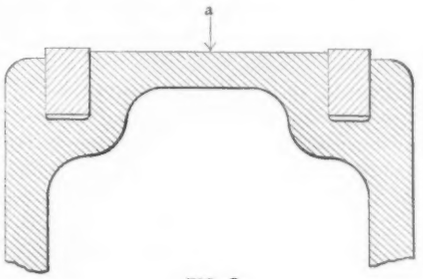


FIG. 1

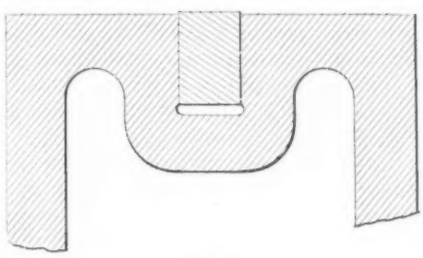


FIG. 2

be somewhat loose sidewise in their grooves, and it would be only fair to suppose that the ring at the high-pressure end of the piston would leak more steam than the one at the exhaust end, on account of difference in pressures.

The average unbalanced steam pressure ought not to be greater on the type of piston shown in Fig. 2 than in Fig. 1. Each piston has the same contact area with its cylinder. It is obvious, from the great diversity of pistons, that a standard would be desirable, then an engineer would need be conversant with but one style of piston. On large horizontal engines a tail-rod is practically a necessity.

The semi-spherical piston-connecting-rod connection for the quintuple gas engine shown on pages 812-13 of the December issue does not seem to be better than the popular cylindrical pin through the piston. The piece marked *A* in Fig. 3, page 813, will soon batter the threads on the piston and then play back and forth. The feather edges on the same piece are objectionable, also. I beg to suggest the design shown in Fig. 3 herewith as being an improvement. All parts of the joint, except the rod, would be rigid when screwed up on paper liners, which of course allows compensation for wear of the spherical surfaces. AMOS PRICE, JR.

Youngstown, O.

Best Location for Back-pressure Valve Weights

Apropos of my article entitled, "A Convenient Method of Weighting and Dismantling a Back-pressure Valve," which appeared in the November issue, page 670, I was one day during that month visited by an engineer friend who carried his right arm in a sling. On inquiring the cause, he informed me that the weight on the lever of his back-pressure valve is located close to the ceiling, in-

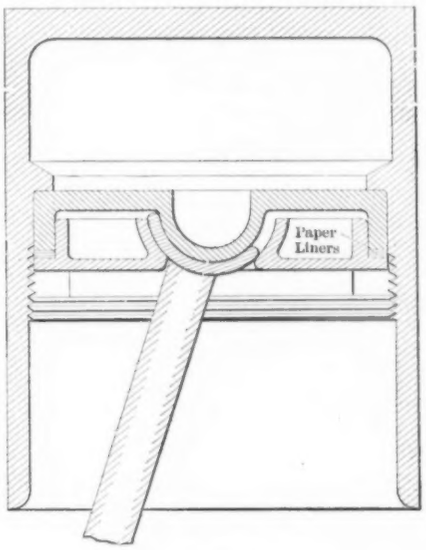


FIG. 3

stead of near the floor, as I had recommended in my article, which, however, he had not seen. The weight fell from the lever upon his hand, resulting in a compound fracture of the forefinger. It was fear of an accident of this kind that prompted me to locate my back-pressure valve weight where it could do no harm.

W. H. WAKEMAN.

New Haven, Conn.

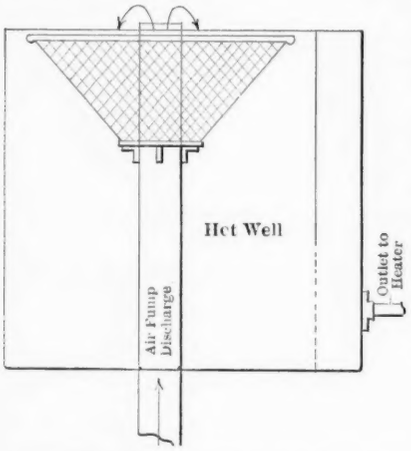
Ingenious Oil Separator for Exhaust Pipe

We have in our plant a 22x42 and 48-inch cross-compound engine which exhausts into a surface condenser. This installation was made without an oil separator in the exhaust pipe and by the time I made the discovery the boilers were suffering. I had to have a sheet "set up" in one of them, and the insurance company served notice that they would not carry the risk unless the oil was taken out of the feed-water; so it was up to the

engineer at once. The discharge pipe from the air pump passed up through the bottom of the hot-well vertically, and it occurred to me that by extending this pipe a little above the top of the hot-well and placing a circular basket of wire netting around the pipe and filling it with straw, I could arrest the oil while the water found its way through the well. This was done. I also substituted a straight mineral cylinder-oil for the compounded oil which had been used up to that time.

This plant had only been in operation six weeks when I took charge of it. The oil separator has been in use over four years and the boilers have never been troubled on account of oil. My experience is that the small percentage of oil which enters them—it being strictly mineral—acts as a scale remover.

Referring to the accompanying sketch, the basket is first lined with burlap and



IMPROVISED OIL SEPARATOR FOR EXHAUST PIPE

then filled with straw. This is changed three times a week, the plant running 143 hours per week. The basket is made of brass wire netting, 1¼-inch mesh, and is supported on brackets bolted to the pipe. It can be slipped off at any time. A partition was placed in the hot-well, the lower part being made of netting, so that if any straw should get into the well it could not get into the pipe leading from the well to the heater.

This separator cost very little to install and seems as good as when first placed, is always open for inspection or for renewing the oil-catching material, and satisfies the insurance company.

Alton, Ill. E. N. GYNERE.

The N. A. S. E. and Engineers' Wages

I read with interest your editorial in December POWER on engineers' wages. Every one of your readers, who is in the least concerned in regard to this wage question, must appreciate the stand you take on the subject. I am curious to know what the leaders in the N. A. S. E. have to say to the fact that their fellow members in New York City, the stronghold of the association, receive comparatively less compensation for their services than engineers in other localities.

The writer is a member of the N. A. S. E., is proud of it, and fully appreciates the benefits educationally and socially open to its members; but any advantage to its members financially seems to be an open question. For some reason or other, the idea that if a man fits himself for a better position he is sure to get it seems to materialize rather slowly in stationary engineering. It is claimed that engineers' wages have been raised since the N. A. S. E. entered the field, and on that account; but wages have also advanced in other trades and even in greater proportion. Is it because our work is not appreciated by the public? I think it is.

This wage question has similarly been up for discussion in *The Engineer*, of Chicago, and I wish to quote from a letter written by an employer which appeared in that paper in December, 1906, as follows:

"Most employers have but a poor conception of the work done in the engine-room and superior results, although involving marked economies, are even less often recognized and appreciated. Dealing with specific cases, the man who can best express himself is the one to be given the first hearing, other things being equal, and the greatest consideration. The means of expression must necessarily be those of the employer to whom sketches, for example, are usually quite unintelligible."

There are apparently only two ways whereby the standard of wages for stationary engineers can be raised: either by the same methods used by labor unions—a proceeding forbidden by the constitution of the N. A. S. E., and equally objectionable to the majority of its members—or by making our services better appreciated and making the employer understand that we are worth somewhat more than common day laborers. The N. A. S. E. fits a member who is willing to profit by his membership to hold a responsible position, but many of us never get the opportunity to prove our fitness.

The N. A. S. E. has a membership of 20,000 engineers, but the public at large does not know much about us, or the purpose for which we are banded together, except when we meet once a year in convention, at which time outsiders are surprised to learn that such a well-behaved, intelligent-looking body of men are really nothing but stationary engineers whom they cannot dissociate in their minds from a bunch of greasy, ill-smelling, overall-wearing individuals always poking round an engine with an oil can and monkey wrench.

The American Society of Mechanical Engineers is known the world over, and a member can print the letters A. S. M. E. on his business card to advantage; it is the same with the A. S. E. E., the A. S. C. E., etc.

There certainly must be some means by which a membership certificate issued by an association built on such sound principles and with such a following as the N. A. S. E. can be made of some financial benefit to its holder. We have as members in the N. A. S. E., active and honorary, men way up in the social and educational world, who have done great work in the past to help their fellow members to become better engineers, as well as better citizens, and judging from the editorial referred to above, I presume it is these men who now have taken up the problem of improving our conditions financially.

In this connection I wish to submit the following: At present when a man applies for membership in our association, he is put through a simple examination to show that he has certain qualifications as an engineer; and if admitted he is entitled to full membership and a certificate. Suppose, instead of this, the certificate was withheld until a member has had time through the educational features of the N. A. S. E., through his own efforts, or both, to qualify for a rigid examination, say something at par with the examination for the best existing first-class state license, and then if the candidate is successful, give him his certificate.

In the meantime educate every employer of engineers, by means of advertisements, circulars or otherwise, to recognize the worthiness of the N. A. S. E. engineer, and the qualifications such a man must possess before he is certified to by the association. The employer will soon understand that the N. A. S. E. certificate on the wall in his engine-room means economical and efficient service, and the time will not be far off when a member of the N. A. S. E. will not only be given the first hearing, but will draw better pay. At the same time I think this would work as a stimulus to many members who are normally ambitious enough, but who get discouraged when they see no material results of their efforts. I would very much like to see this subject discussed by readers of POWER who are better qualified than the writer to judge of its merits.

Chicago, Ill. R. CEDERBLOM.

"How One City Plant Is Run"

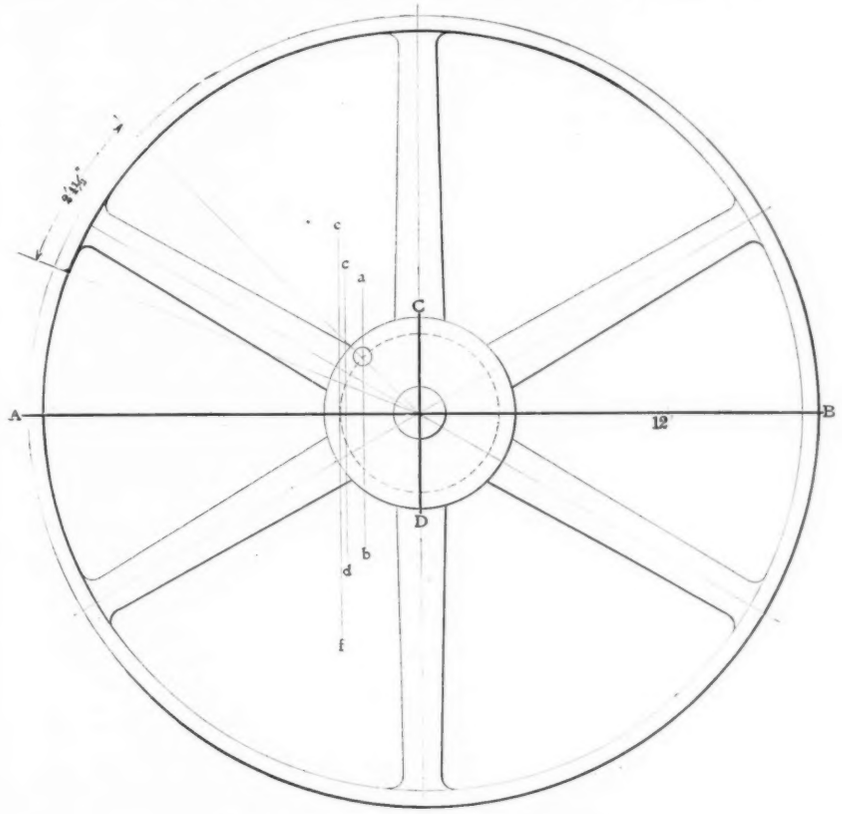
In response to J. Green's letter on page 62, January POWER, I would state that the old and new boilers are rated alike, for 150 horse-power, each containing 4-inch tubes of the same length. The new boilers are hung from a steel frame, and the old are supported on lugs in the wall. The old boilers were provided with domes, but the new boilers have neither domes nor (I have been told) dry-pipes, although I cannot positively say as to the

latter. Now my brother engineer can decide for himself whether the gain is worth the $1800 referred to in my letter in the November issue. A. SUBSCRIBER.

Perry, Okla.

Effect of Compression

Until quite recently I had accepted as truthful the old philosopher's statement that "there is nothing new under the sun," but I have been compelled to change my mind by recently coming into collision with an expert (?), who claims that in a tandem-compound engine "the proper place for compression is the high-pressure cylinder." That the effects of compression, the benefits derived from a proper amount of it and the reason why too much is wasteful, and consequently undesirable, are not as well understood as they should be by engineers, is evidenced by the many freak reproductions we see in our trade papers.



ILLUSTRATING "EFFECT OF COMPRESSION"

I have undertaken in the accompanying diagram to show graphically the effects of compression. *A—B* is the center line of the engine; *C—D* is a line drawn vertically across the crank-disk through the center of the main shaft; *a—b*, a line drawn through the crank-pin on the lower head eighth vertically; *c—d* is a line drawn likewise through the crank-pin on the sixteenth, and *e—f* is a vertical line drawn through the crank-pin on the head center.

Were it not for the error due to the angularity of the connecting-rod the distance from *C—D* to *a—b* would be equal to the travel of the piston, while the crank-pin is moving through the distance between these two lines; the distance between the lines *a—b* and *c—d* would be equal to the travel of the piston while the crank-pin is moving the distance between these two lines, or from the eighth to the sixteenth; while from *c—d* to *e—f* equals the travel of the piston while the crank-pin moves from the sixteenth to the head center.

Now, the force which must be depended upon mainly to carry the engine over the centers or dead points is in the fly-wheel, and the place where this force is most effective is in the rim. We will assume that the engine we are considering has a stroke of 36 inches, and has a fly-wheel 12 feet in diameter, the circumference at the rim being approximately 38 feet (37 feet 9 inches). We will then, to make the matter clear, assume that the force *is* in the fly-wheel rim (and most of it is). We know that compression acts as a resistance to that force, and its *effect* depends upon the *relative distances through which the force and resistance are acting* in a given time.

Without being absolutely exact, the diagram shows clearly that while the crank-pin is passing from the sixteenth to the center, the force in the rim of the fly-wheel moves through a distance of 2 feet 4½ inches, and while the crank-pin moves from the eighth to the sixteenth, or from *a—b* to *c—d*, the force acts through the same distance as before, but the resistance acts through a distance about *four times as great* as while the crank-pin is moving from *c—d* to *e—f*. Hence, four times as much energy is required to effect compression through the distance *a—b* to *c—d* as would be required to effect it through the distance *c—d* to *e—f*, due to this cause alone.

It should be perfectly clear then that there is a point beyond which we cannot go without putting an *effective brake* on the engine and paying a dear price for what we save by compression. But within proper limits compression is a saving, because as the crank-pin approaches the centers the crosshead acts very much as a toggle joint and gives the forces acting against the resistance due to compression greater leverage than they would have earlier in the stroke.

That a proper amount of compression is an economy as well as a benefit and a necessity in other ways is pretty nearly universally admitted. Then why not let each cylinder accomplish this saving and thus endeavor to equalize the work of compression in both cylinders, as well as in other respects, whether it be a tandem or cross-compound? If we put all the compression on the high-pressure piston required to secure smooth running, we will have too much, while late exhaust closure and little compression in the low-pressure cylinder is causing waste where it is most telling on the economical performance of the engine.

The expert who recommends such freakish practice is at variance with all designers and builders of compound engines, as they invariably advise earlier exhaust closure on the low-pressure cylinder than on the high-pressure.

Again, too little compression means, generally, not enough lap to protect the working edges of the valve and ports properly, and they are more likely to become "steam cut" and leaky than if proper lap is given, and if little or no compression curve shows on the indicator diagram, no one but an "expert" would be likely to notice it when it did occur.

WILLIAM WESTERFIELD.

Charleston, Mo.

Referring to the letter by William Westerfield, I would say that in providing for compression the inertia of the wheel rim should have nothing to do with it. The engine would go over the center itself without a wheel. The inertia of the wheel rim carries the load while the engine is passing the center, and during light loads after the expansion has reduced the pressure in the cylinder to a low point.

The only use for compression is to stop the reciprocating parts, and the amount required will depend upon the weight of these parts and the number of revolutions. The reciprocating parts of a tandem engine are much heavier than those of a single engine, and a high rotative speed will require more compression than a single engine.

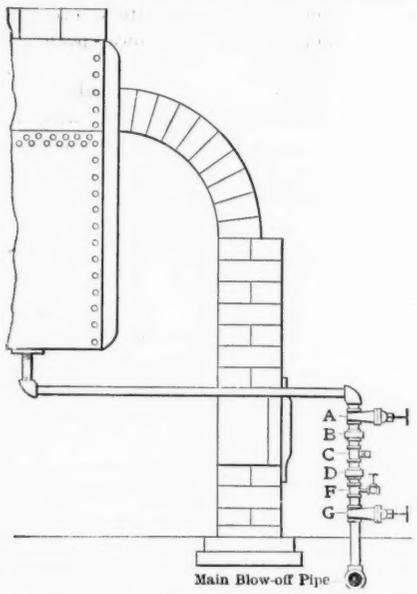
The area of the high-pressure piston is about one-fourth that of the low-pressure piston, and if the high were depended upon to receive all the compression it would be necessarily very high. With the same engine and the same cut-off, when the compression is high the cut-off corner will be rounded and the terminal pressure will be higher, thus losing something in economy, and this could better be afforded in the high-pressure than in the low-pressure; but a small amount of compression will do as much service in the low-pressure as a high compression in the high-pressure cylinder. These two items are the chief things to be considered with a reasonable amount of compression. An unreasonable amount of compression is when it is carried to the point that it holds back on the wheel.

The man who gives compression sufficient to check the wheel in any way is using coal to take energy out of the wheel, which has already cost him good money to have put in. W. E. CRANE.

Waterbury, Conn.

Good for the Maker of Fittings

In the issue of September last, page 571, Mr. Jahnke pointed out the error of connecting two or more boilers to one blow-off pipe and providing only one valve or cock to each boiler; because, first, of the impossibility of detecting which one leaks, and, second, the difficulty of repairing the leak, with the boilers running. Now, herewith is a good scheme for piping off the boilers, which, though a little more expensive, will pay in the end and has the advantage of great facility for repairs.



MR. GRIGGS' SCHEME FOR PIPING BOILER

Next to the boiler place the gate-valve *A*, then the union *B;* next the ground cock *C*, then the union *D*, then the tee *F* with one-half-inch branch, to which is connected a valve; then insert the gate-valve *G*, and finally a nipple connection to the main blow-off pipe. We depend for a tight job on the ground cock *C*, the gate-valves serving to cut off the pressure from both sides while the boiler is running and allowing the use of the main blow-off pipe at any time without causing inconvenience. The unions make it an easy job to disconnect the ground cock By having *C* and *G* shut and opening *F*, it is very easy to ascertain whether the cock *C* leaks. It will well repay the expense and trouble of connecting up all these fittings, as it is so easy for the engineer to keep everything perfectly tight, and the slightest leak in a blow-off is very wasteful. EUGENE L. GRIGGS.

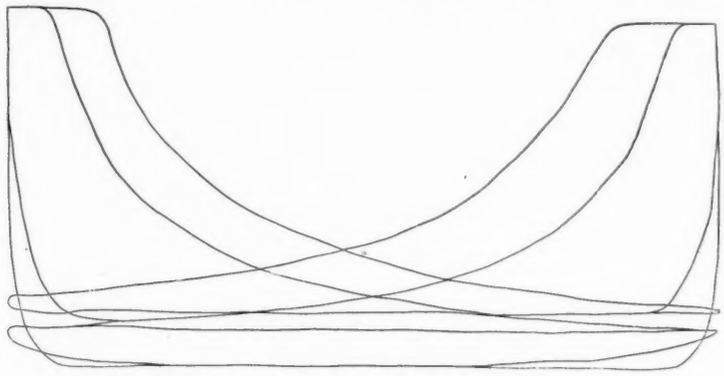
Allston, Mass.

The Effect of Using a Condenser

I inclose a diagram that I have taken with and without the Baragwanath condenser in use. I think the cards will speak for themselves. I am carrying 27 inches vacuum easily, and I must say I don't believe the old book that said a condenser on a simple engine would not pay, for I found a saving of 600 pounds of coal a day.

My little plant consists of one 16-foot x 60-inch boiler, with 48 4-inch flues, a 14 x 36, 85-revolution Cooper-Corliss engine, a Snyder-Hughes 10x7x10 pump for general service, supplying 20 power-houses besides the foundry and nickel-plating room. The load runs from 70 to 140 horse-power during the day, and I am burning less than a ton of coal in twenty-four hours. The shop runs ten pumps under steam all the time, but does little at night.

Elyria, Ohio. A. N. WALKER.



CARDS TAKEN WITH AND WITHOUT CONDENSER IN USE

Comment on "The Steam Plant of the Wood Worsted Mill"

While reading the article in February POWER on "The Steam Plant of the Wood Worsted Mill" I was surprised to note that the boilers are horizontal-tubulars, 78 inches diameter, of 7/16-inch plate of 60,000 pounds tensile strength. This would give a factor of safety of 4, only, which I think is altogether too low for such boilers carrying 160 pounds steam pressure. Boilers in use show wear and will not last long enough with such a small factor of safety.

I should have preferred water-tube boilers at an increased cost of, perhaps, three dollars per horse-power, as the saving in the cost of building would have been in their favor. I know all do not think alike, however.

Mr. Diman has attained some remarkably good arrangements in this work, and I write not so much in the spirit of criticism as with a desire to obtain information.

I would like to ask what formula builders use for boilers to obtain the proper thickness of metal in tubes and flues. I find that some take into consideration the length of the tube, while others do not. R. C. W.

Somerville, Mass.

"Why the Difference in the Steam Line?"

On page 115 of February POWER, Mr. Ackerman's low-pressure diagrams are to my mind similar to the cards shown on page 794 of the December issue, and I believe that if Mr. Ackerman will take the time and look up Chas. J. Mason's explanation, he may find the reason for the low initial pressure at the head end. I notice the atmospheric line is missing. The cards from the high-pressure cylinder are very nearly alike, and evidently the engine was running with a light load when the cards were taken, as the cut-off is early and the expansion curve falls below the back-pressure line.

The head-end diagram of the low-pressure cylinder shows there is not enough compression, as compared with the other cards. As the crank-end diagram is fairly good, I believe that the trouble may have been caused by the indicator piping. Perhaps the indicator cock was not opened all the way, or the little globe valve at the head end of the cylinder. This will always make a difference in the steam line. Or, again, it could be that the piston of the engine traveled over the hole at the head end, into which the indicator piping is screwed. JOHN I. BAKER.

Allentown, Pa.

First Exposition of Safety Devices Nucleus of an American Museum of Security

Rarely has a project for social and industrial betterment been launched under conditions as favorable as those which attended the first exposition in this country of safety devices and industrial hygiene, which was opened January 29 last in the Museum of Natural History, New York, under the auspices of the American Institute of Social Service. The immediate object of the exhibition, which was international in scope as far as the exhibits were concerned, was to direct the attention of American public opinion to the necessity of doing something to lessen the causes of accident in and outside of industrial establishments, by means of a permanent museum of safety devices, where all problems of safeguarding life and limb may be studied in their working details.

The movement had received a decided impetus from the last census report, which showed that there were upward of 57,000 violent deaths in the United States in 1900, and where figures for comparison were available it appeared that there were twice as many accidents in this country, relatively, as in Europe. This, together with the fact that in Europe, and even in Russia, there are such museums, designated "museums of security," which coupled with workingmen's associations, government aid and sympathetic indorsement by individuals and societies, have conserved public safety to a remarkable extent, pointed the way to the achievement of a long-cherished purpose in this country.

Invitations were extended to manufacturers of practical safety devices the world over, as well as to directors of museums of security and affiliated organizations, to participate in the exposition, with spontaneous results which far exceeded anticipation. In fact, it is announced that what with guarantees in hand, not only is the establishment of an American Museum of Security assured, but it will start with a far greater number of permanent exhibits than did the Berlin and Amsterdam museums, the former of which is cited as the model upon which the American museum will be based. More than one hundred and twenty firms, associations and individuals accepted the invitation, and there were nearly one hundred exhibits in place when the exposition opened.

In accepting exhibits care was taken to exclude what seemed to be "impossible" devices. It was also aimed to have the exposition comprise "live exhibits," i.e., machines or safety appliances in operation, models of actual or reduced size, and photographs. There were also shown reports and documents relating to the betterment of workingmen's conditions, and last but not least the Red Cross Society's and other first-aid-to-the-injured outfits.

Among representative American firms in the power-plant field who contributed to the success of the exposition are the Westinghouse Companies, General Electric Company, Consolidated Engine Stop Company, Northern Specialty Company, H. & M. Valve Company, New York Edison Company, the Lunkenheimer Company. The exhibit was open daily, Sundays and evenings included, for the better part of three weeks, and it was visited by thousands of persons. Its inauguration and the demonstration of its practicability were directly in charge of Dr. Josiah Strong, president of the American Institute of Social Service, and Dr. William H. Tolman, director, assisted by Mrs. Tolman and Miss E. G. Symonds, of the Institute staff.

During the exposition Dr. Tolman, who, with Mrs. Tolman, has spent the past nine years personally investigating such matters at home and abroad, gave several illustrated lectures, and on Saturday, February 9, addressed a large body of public school children, who were present with their teachers by invitation.

Aside from the expectation that in addition to philanthropists and humanitarians now interested others will become encouraged to aid the movement, it is anticipated that the State will render aid, especially in view of the unequivocal indorsement of such a movement by Governor Hughes at a banquet at the Waldorf-Astoria hotel on the eve of the opening of the exposition.

Incidental to the plans so far perfected it is proposed to establish in connection with the museum a laboratory for inventors, where supplies may be purchased at cost and inventions developed, to be passed upon by a jury of experts, after which accepted models will be placed on exhibition for inspection by manufacturers, capitalists and others.

At the close of the exposition each exhibitor was presented a handsome commemorative medal, furnished by the Gorham Company, which is reproduced herewith.

COMMEMORATIVE MEDAL PRESENTED TO EXHIBITORS AT THE INTERNATIONAL EXPOSITION OF SAFETY DEVICES AT NEW YORK

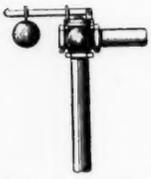
A Central Machinery Exchange at Boston, Mass.

There was recently opened in the Oliver Building, corner of Milk and Oliver streets, Boston, Mass., a machinery exchange, in accordance with a project originated by Elroy N. Heath, New England representative of the H. B. Smith Machine Company. The plan is somewhat after the style of the Philadelphia Bourse; on the first floor of a modern office building space is assigned to one firm in each line of a variety of machinery and supplies. Numerous facilities are provided for the expedition of business and the comfort of the tenants. The building is always open.

About twenty firms have located their New England offices and exhibits in the exchange. Bulletins are mailed regularly by the management to 10,000 machinery users in New England, advertising the names and products of exchange members. It is believed that the new venture will result in economy of time, effort and expense for buyers and sellers alike.

Recently forty members of the Washington Society of Engineers visited the proposed Berning turbine plant of the Potomac Electric Power Company, now under construction. This plant will cost $1,000,000 and will be completed in four years. It will be equipped with a 5000-horse-power turbine.

THE SAFETY VALVE



THE STUDY of the safety valve has been the first step of many a man in scientific engineering. Induced to its study by the necessity of solving its problems before the examiner, his consideration of this simple device has led him into the computation of areas, into a study of the principle of the lever, of moments of forces, of the velocity of flow of steam and other fundamental principles of mechanics. This applies to those who have studied the subject intelligently, not to those who have attempted to get over it by learning a rule by rote, simply to be confounded when confronted by another rule, or a case to which their rule would not apply. The whole subject is so simple that an hour's study will put a man in possession of the fundamentals so that he can make his own rules or solve any problem without a rule, from a sheer understanding of the principles involved.

PRESSURE PER SQUARE INCH

A CUBIC foot of water weighs, in round numbers, 62 pounds. If you can imagine ten cubic feet packed one above the other, as in Fig. 1, they would make a column weighing some 620 pounds, supported on a base one foot square, so that the pressure would be 620 pounds *per square foot.* The water in a tank or pond may be conceived to be divided into columns of this kind, and it will be seen that there will be a pressure on the bottom of 62 pounds per square *foot* for every foot of depth. But, in the square foot supporting this weight there are 144 square inches; and as the pressure is evenly distributed, each square inch carries:

$$62 \div 144 = 0.43 \textit{ of a pound}$$

for each foot in depth, and the pressure in the case of the column ten feet in hight would be 620 pounds per square foot, or 4.3 pounds *per square inch.*

Just as the tank or pond could be conceived to be divided into columns of one square foot section, each square foot can be conceived to be divided into 144 columns of one square inch section, as shown in Fig. 1, and each foot in hight of such column like the piece marked *A* would weigh $\frac{1}{144}$ of the whole weight of the cubic foot of which it is the $\frac{1}{144}$ part, and press upon its square inch of base with a pressure of:

$$62 \div 144 = 0.43 \textit{ of a pound.}$$

As this pressure in a liquid or gas is exerted in all directions, it is evident that the pressure on the horizontal piston in Fig. 2 would be 4.3 pounds *per square inch*, and if it has an area of 30 square inches there would be a force of:

$$4.3 \times 30 = 129 \textit{ pounds}$$

forcing the piston to the right; and since there is at a depth of 9 feet a pressure of 3.87 pounds *per square inch* the valve at the left would have 3.87 pounds pushing upward on *each square inch* of its exposed area, i.e., the area corresponding with the diameter *a b*, and if that area were 30 square inches it would take:

$$30 \times 3.87 = 116.1 \textit{ pounds}$$

to hold the valve closed against that pressure.

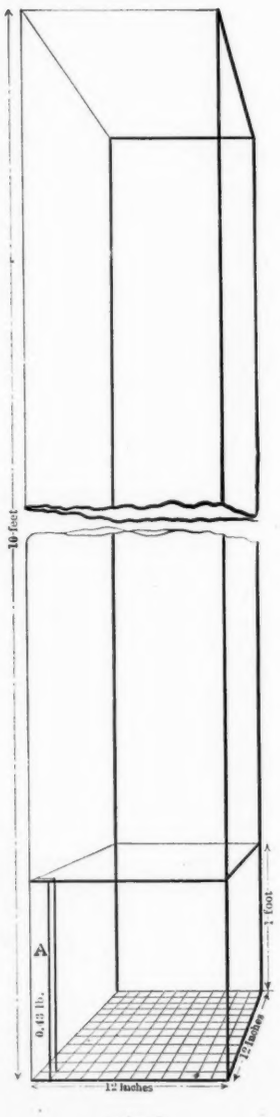


FIG. 1

The steam gage shows the pressure *per square inch.* If the gage points to the 100 mark it indicates that if the pressure existing in the boiler were exerted upon one square inch of area, Fig. 3, it would push with a force of one hundred pounds. If exerted upon an area of one-half a square inch, Fig. 4, it would push with a force of fifty pounds; upon an area one-half inch square, or one-quarter of a square inch, Fig. 5, twenty-five pounds; upon an area of one square foot, or 144 square inches, 14,000 pounds, etc.

The force exerted by the steam to lift a safety valve depends then upon the area of the valve as well as upon the intensity of the pressure.

TO FIND THE AREA OF A CIRCLE

The area of a one-inch circle is 0.7854 of a square inch, the difference, 0.2146, between this and the full square of the diameter being taken up by the corners, Fig. 6. If the side of the square is *doubled* the area of the square will be multipled by *four*, as is plainly shown by Fig. 7, which obviously contains *four* squares of the area of that shown in Fig. 6, although its side is but *twice* as long; and it is equally evident that the inclosed circle bears the same proportion to the total area in both cases and that the shaded area of the circle in Fig. 7 is *four* times that in Fig. 6, although its diameter is but *twice* that of the smaller circle. If we *treble* the length of the side the area of the square will be multiplied by *nine*, always the *square* of the side, i.e., the side multiplied by itself.

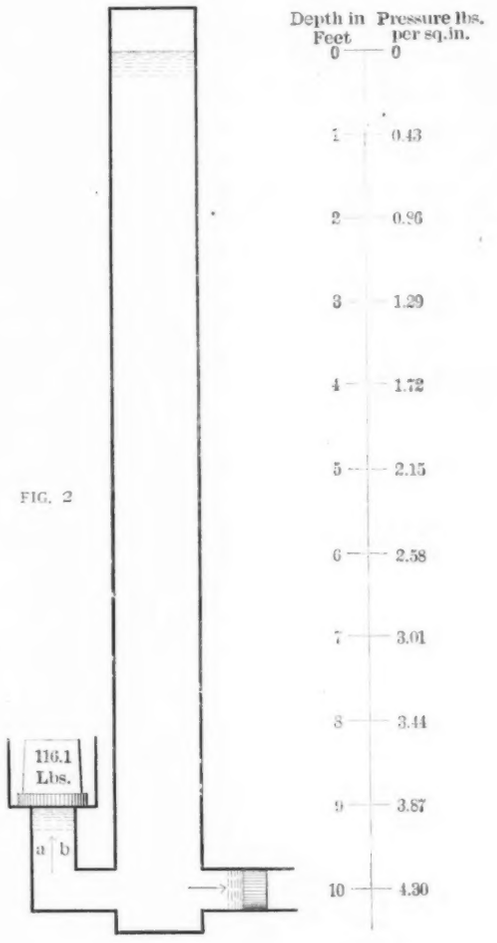


FIG. 2

The area of any circle may be found by multiplying the area of a one-inch circle (0.7854 square inch) by the square of the given diameter.

In Fig. 7 the diameter is two inches and the area is:

$$2 \times 2 \times 0.7854 = 3.1416 \text{ square inches.}$$

The area of a four-inch circle would be:

$$4 \times 4 \times 0.7854 = 12.5664 \text{ square inches.}$$

It may aid in remembering the factor 0.7854 to know that it is one-fourth of 3.1416, the number by which the diameter is multiplied to get the circumference.

The area of a triangle is obviously one-half the product of its base and hight. In Fig. 5a the product of the base $A\,B$ and the hight $A\,D$ would be the area of the rectangle $A\,B\,C\,D$ and the shaded area of the triangle is obviously one-half of this, for the two unshaded portions put together would make a similar triangle. This is just as true if the base is an arc of a circle as in Fig. 5b, and just as true if the base incloses the apex as in Fig. 5c. The circle is therefore a triangle with a circular base 3.1416 times the diameter or 2×3.1416 times the radius, and with a hight equal to the radius, and its area (one-half the product of the hight and base) is:

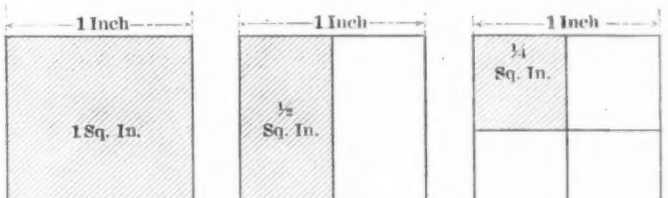


FIG. 3 FIG. 4 FIG. 5

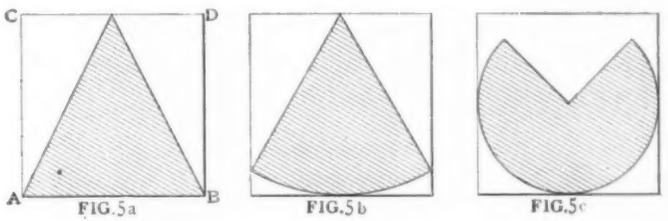


FIG. 5a FIG. 5b FIG. 5c

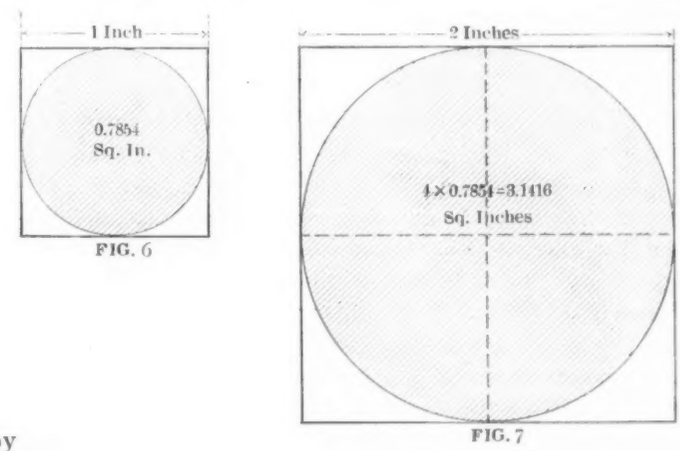


FIG. 6 FIG. 7

$$\text{Area} = \frac{\text{radius} \times 2 \times 3.1416 \times \text{radius}}{2} = 3.1416 \text{ radius}^2,$$

so that the area equals 3.1416 times the square of the radius, and since the radius is one-*half* the diameter the square of the radius is the square of the diameter divided by *four*:

$$\text{Area} = 3.1416\, r^2 = 3.1416 \frac{D^2}{4} = D^2 \times \frac{3.1416}{4} = 0.7854\, D^2.$$

EFFECT OF PRESSURE IN LIFTING A VALVE

Suppose the 3-inch valve in Fig. 8 to be loaded with six weights of 100 pounds each and that the valve and steam weighed 30 pounds, what would the pressure per square inch have to be to lift it?

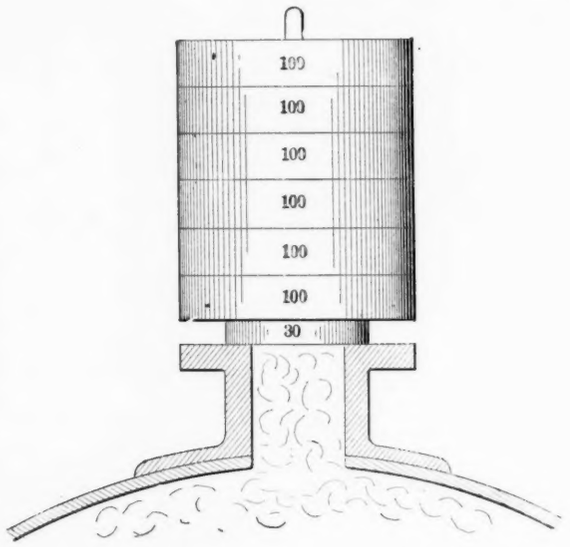


FIG. 8

The total weight to be lifted is 630 pounds. The total upward pressure must equal this, and if 630 pounds is exerted on 7.0686 square inches (the area of a 3-inch valve, see table) the pressure on each square inch will be:

$$630 \div 7.0686 = 89.1 \textit{ pounds.}$$

How much load would have to be put upon the same valve to allow it to blow off at 75 pounds per square inch?

If the pressure exerts 75 pounds on one square inch it would exert on the 7.0686 square inches of the valve which is exposed to it:

$$75 \times 7.0686 = 530 \textit{ pounds,}$$

which must be the combined weight of the valve and the weights with which it is loaded.

Fig. 8 does not show a practicable valve, but is sufficient to illustrate the point that the force tending to lift the valve must equal that holding it to its seat (in this case the dead weight of the valve itself and the weights with which it is loaded), and that this upwardly acting force is the area of the valve in square inches, multiplied by the pressure per square inch. Such a dead-weight valve is ponderous and impracticable and the usual practice is to use a lighter weight, increasing its effect by leverage, or to hold the valve to its seat with a spring.

THE PRINCIPLE OF THE LEVER

Suppose a strip of board balanced over a sharp edge as in Fig. 9. If equal weights be placed upon it at equal distances from the center it will still be in balance. If one of the weights be moved in *half* of the distance to the point at which they are balanced, as in Fig. 10, the other weight will have to be *halved* to preserve the equilibrium. If one of the weights be moved to *one-third* of its distance from the balancing point, as in Fig. 11, the other weight will have to be reduced to *one-third* of its original magnitude to preserve the balance at the original distance.

Notice that in each case the product of a weight by its distance from the point over which they balance is the same as the product of the weight which balances it and its distance from the same point. Suppose the weights in Fig. 9 to be each 20 pounds, each at 12 inches from the center. Here obviously the weights and distances being the same their products are equal:

$$20 \times 12 = 240 \quad \textit{and} \quad 20 \times 12 = 240.$$

When the right-hand weight is moved in to 6 inches from the center the other had to be reduced to 10 pounds:

$$10 \times 12 = 120 \quad \textit{and} \quad 20 \times 6 = 120.$$

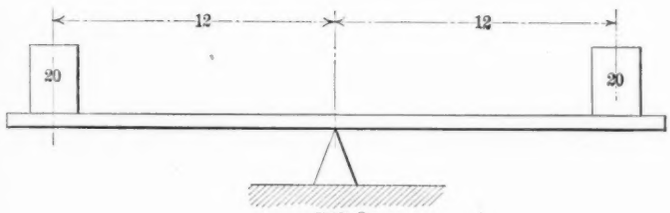


FIG. 9

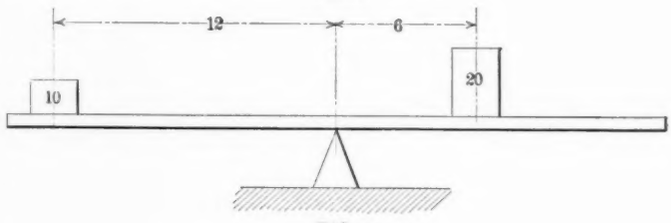


FIG. 10

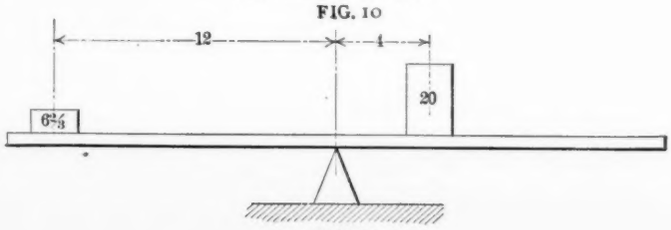


FIG. 11

When the left-hand weight was moved in to 4 inches from the center the other had to be reduced to 6⅔:

$$6\tfrac{2}{3} \times 12 = 80 \qquad \textit{and} \qquad 20 \times 4 = 80.$$

The same principle applies in Fig. 12, where the force exerted by the man, multiplied by the distance AB, must, if he lifts the machine, equal the pressure with which the load bears on the bar at the point C, multiplied by the distance BC of that point from the point B around which the lever turns. In mechanics, this point, the B of Fig. 12, is called the "fulcrum" and the product of the load, weight or force by its distance from the fulcrum is called its "moment." In the case described by Fig. 9 the moment of each weight is 240; in that of Fig. 10, 120; in that of Fig. 11, 80; in that shown in Fig. 12 the moment of the load is the weight or force with which the load bears on the point C, multiplied by its distance from the fulcrum B, and the moment of the force is the force which the man exerts upon the bar at A, multiplied by the distance of that point from the fulcrum.

Notice that in Fig. 13 the fulcrum is at one end of the lever instead of between the load and force as in the other examples. The principle is the same. The fulcrum is the *stationary* point about which the load and the force move. In Figs. 12 and 13 it is evident that the shorter the distance between the load and the fulcrum the less the man will have to exert himself.

The point to grasp and remember is that the *moments must be equal* in order for the force to balance or lift the load.

Equal Moments Produce Equilibrium.

There are four important things about a lever:

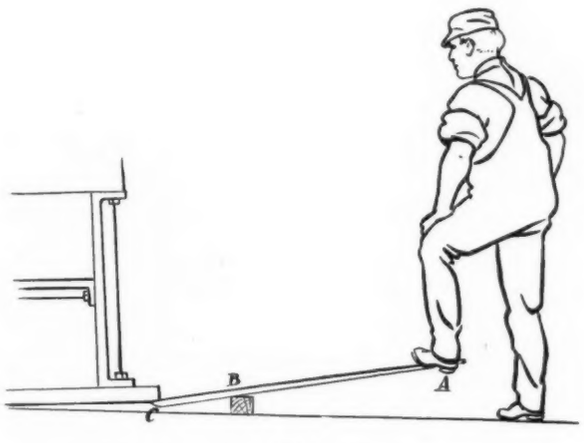


FIG. 12

L = the load,
F = the force applied to balance or overcome the load.
D_l = distance of the load from the fulcrum,
D_f = distance of the force from the fulcrum.

If any three of these are known the third can be easily determined, for as has been just explained,

Force × distance of force = load × distance of load.

$$F \times D_f = L \times D_l.$$

Moment of force = moment of load.

TO FIND THE FORCE REQUIRED TO LIFT A GIVEN LOAD:

FORMULA:

$$F = \frac{L \times D_l}{D_f}$$

FIG. 13

RULE—*Multiply the load by its distance from the fulcrum, and divide by the distance at which the force is applied from the fulcrum.*

TO FIND THE DISTANCE AT WHICH A GIVEN FORCE MUST BE APPLIED FROM THE FULCRUM TO BALANCE A GIVEN LOAD:

FORMULA:

$$D_f = \frac{L \times D_l}{F}$$

RULE—*Multiply the load by its distance from the fulcrum and divide by the given force.*

TO FIND THE LOAD WHICH MAY BE LIFTED WITH A GIVEN FORCE:

FORMULA:

$$L = \frac{F \times D_f}{D_l}$$

RULE—*Multiply the given force by the distance of its point of application from the fulcrum and divide by the distance of the load from the fulcrum.*

TO FIND THE DISTANCE AT WHICH A GIVEN WEIGHT OR LOAD MUST BE PLACED FROM THE FULCRUM TO BALANCE A GIVEN FORCE:

FORMULA:

$$D_l = \frac{F \times D_f}{L}$$

RULE—*Multiply the given force by the distance of its point of application from the fulcrum and divide by the load.*

THE LEVER SAFETY VALVE

Effect of the Leverage of the Ball

Suppose the weight instead of setting directly upon the valve, as in Fig. 8, is applied through a lever, as in Fig. 14. From what has preceded it will easily be seen that the weight multiplied by its distance from the fulcrum will equal the force which it will exert upon the valve stem multiplied by the distance of its point of application from the fulcrum.

Weight of ball	×	Distance of ball from fulcrum	=	Pressure of ball on stem	×	Distance of stem from fulcrum

Let the weight equal 75 pounds,
distance of weight from fulcrum 32 inches,
distance of stem from fulcrum 2¾ inches,

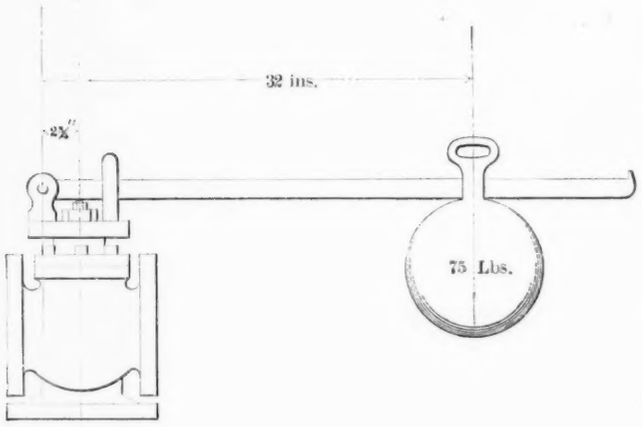


FIG. 14

what will be the force exerted by the ball to hold the valve to its seat?

$$\frac{\text{Weight of ball} \times \text{Distance of ball from fulcrum}}{\text{Distance of stem from fulcrum}} = \text{Pressure of ball on stem.}$$

Then the moment of the ball is:

$$75 \times 32 = 2400 \textit{ inch-pounds},$$

and:

$$2400 \div 2.75 = 872.727 \textit{ pounds}$$

will be the pressure on the valve stem due to the ball and the moments will be equal:

$$75 \times 32 = 2400 \quad \textit{and} \quad 872.727 \times 2.75 = 2400.$$

Suppose this to be a 4-inch valve, the area of which is 12.5666 square inches. The pressure per square inch upon the under side of the valve necessary to balance the effect of the ball would be:

$$872.727 \div 12.5666 = 69.4 \textit{ pounds}.$$

This is the pressure at which the valve would blow off if nothing but the ball were holding it to its seat. It takes a little additional pressure to lift the valve and to overcome the weight of the lever, as will be explained later, but this is a comparatively small affair and in usual approximate calculations is not taken into account. Neglecting these we can make the following simple

RULES FOR LEVER SAFETY VALVE, NEGLECTING WEIGHT OF VALVE, STEM AND LEVER:

Let W = weight of the ball,
D = distance of ball from fulcrum,
A = area of valve in sq. ins.,
d = distance of stem from fulcrum,
P = pressure per square inch on valve which will balance ball.

TO DETERMINE THE PRESSURE ON A VALVE OF GIVEN DIAMETER REQUIRED TO BALANCE A BALL OF GIVEN WEIGHT AT A GIVEN DISTANCE FROM THE FULCRUM:

FORMULA:

$$P = \frac{W \times D}{A \times d}$$

RULE—*Multiply the weight of the ball by its distance from the fulcrum. Multiply the area of the valve in square inches by the distance of its stem from the fulcrum. Divide the first product by the second and the quotient will be the pressure per square inch required to overcome the weight of the ball.*

EXAMPLE—The stem of a four-inch safety valve s 2¾ inches from the fulcrum. Supposing the valve will blow when the gage shows 7 pounds without any weight upon the lever (i.e., that it takes 7 pounds per square inch on the area of the valve to overcome its own weight, that of the stem and the bearing effect of the empty lever), at what pressure would it blow with a weight of 75 pounds (Fig. 14) 32 inches from the fulcrum?

BY THE FORMULA:

$$P \frac{W \times D}{A \times d} = \frac{75 \times 32}{12.5666 \times 2.75} = 69.4 + 7 = 76.4 \textit{ lbs.}$$

BY THE RULE:

Area of valve	12.5666	75	Weight of ball
Distance of stem	2.75	32	Distance of ball
	628330	150	
	879662	225	
	251332		
	34.558150) 2400 00 (	69.4 *lbs.*	

This is the pressure required to lift the ball. Adding the 2 pounds required to blow the valve without the ball, the answer would be 76.4 pounds. Scratching out the last three figures of the first product saves handling large numbers and does not materially affect the result. If we called this 34.6 (nearer right than 34.5 because the 58 rejected is over one-half) the quotient would still be 69.36.

TO FIND THE WEIGHT REQUIRED TO HOLD A GIVEN PRESSURE ON A GIVEN VALVE:

FORMULA:

$$W = \frac{A \times P \times d}{D}$$

RULE—*Multiply the area by the pressure and by the distance of the stem from the fulcrum and divide by the distance of the ball from the fulcrum. The quotient will be the weight of ball required to balance the steam pressure on the valve.*

EXAMPLE—What weight of ball would be required to allow the valve in the above example to blow off at 80 pounds?

The ball must provide for 73 pounds per square inch, the lever valve and stem taking care of the other two, so that $P = 73$ pounds.

BY THE FORMULA:

$$W = \frac{A \times P \times d}{D} = \frac{12.5666 \times 73 \times 2.75}{32} = 78.8 \text{ lbs.}$$

BY THE RULE:

12.5666	Area
73	Pressure
37699 8	
879662	
917.3618	
2.75	Distance of stem
45868090	
64215326	
18347236	

Distance of ball, 32) 2522.744950 (78.8 *lbs.*

TO FIND THE POSITION OF THE WEIGHT IN ORDER THAT IT MAY EXERT A GIVEN PRESSURE ON THE STEM:

FORMULA:

$$D = \frac{A \times P \times d}{W}$$

RULE—*Multiply the area by the pressure and by the distance of the stem from the fulcrum and divide by the weight of the ball. The quotient will be the distance at which the ball must be from the fulcrum in order to produce a given pressure on the stem.*

EXAMPLE—If the original 75-pound weight had been used, at what distance from the fulcrum would it have had to have been placed to have allowed the valve to blow off at 80 pounds?

BY THE FORMULA:

$$D = \frac{A \times P \times d}{W} = \frac{12.566 \times 73 \times 2.75}{75} = 33.6 \text{ inches.}$$

BY THE RULE—The product of the factor in the numerator is 2695.5 as before and dividing this by 75, the weight of the ball:

75) 2522.74495 (33.6 *inches.*

These simple rules will serve all practical purposes, especially if it is borne in mind that P represents the pressure with which the *ball only* bears upon the stem, not including the weight of the valve, lever, etc., and an allowance be made for these other effects as has been done in the examples. A general idea of what the pressure per square inch required to lift the valve, stem and lever may be is given in column 8 of the table on page 201. It is well, however, to know how to make these allowances accurately, and they will now be considered.

Effect of the Weight of the Valve and Stem

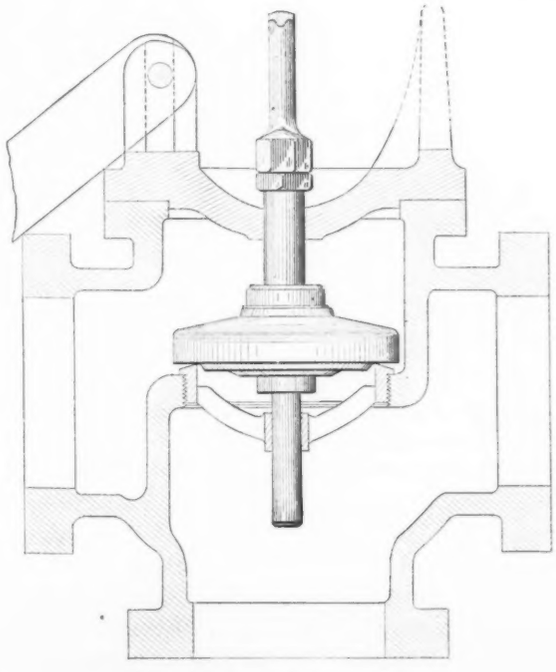


FIG. 15

The pressure acts directly upon the valve and stem without leverage, and must exert a force to balance their weight equal simply to that weight, just as was the case in Fig. 8.

Suppose the valve and stem of a 3-inch valve to weigh 1.5 pounds, how much pressure per square inch would be required to lift the valve from its seat?

Comparing Figs. 8 and 15, it will be seen that this case is the same as the first example given in describing the earlier cut. The total pressure on the valve must be 1.5 pounds, and if 1.5 pounds is to be exerted on 7.0686 square inches, the pressure per square inch will be:

$$1.5 \div 7.0686 = 0.212 \text{ pound.}$$

Column 3 of the table on page 199 gives roughly the weights of valve and stem used on valves of the standard diameters of three makers, and in connection with column 4, which gives the pressure per square inch required to lift the valves of the given weights, serves to indicate the relative importance of this factor of the problem.

The Effect of the Lever

The weight of the lever tends to hold the valve upon its seat. It is evident that it would take a considerable pull to lift the lever of a large safety valve with a cord attached at the point at which the pin bears, as in Fig. 16, and this pull as measured upon a scale would be the force which the valve would have to exert to push the lever up. Every successive particle in the length of the

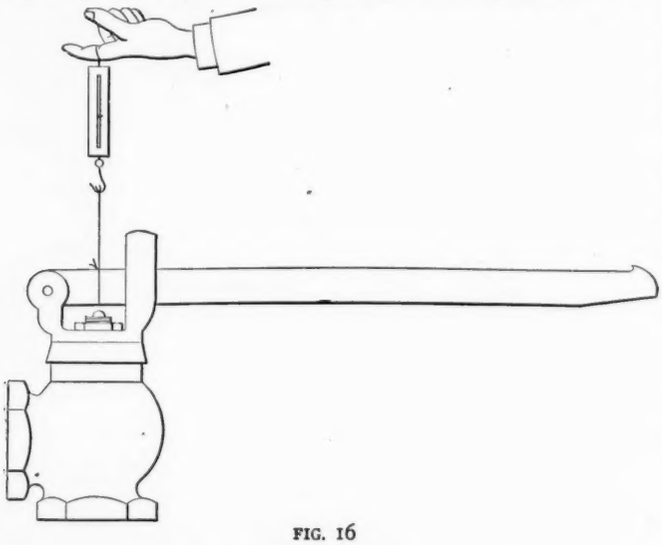


FIG. 16

lever is acting with a different leverage, so that it would at first appear a complicated process to calculate this force; but a body acts in this respect just as though its whole mass were concentrated at its center of gravity, and this makes the problem very simple.

If the lever be taken off and balanced over an edge, as in Fig. 17, the center of gravity will be at the point above the knife edge when the lever is balanced, and the effect of the lever would be the same as if all the mass were concentrated at that point.

Now find the distance of the center of gravity from the fulcrum, from the point around which the lever turns. This will be from the center of the hole when it turns upon a pin, as in Fig. 18, or from the point where it bears if a knife edge is used, as in Fig. 19; the distance *a c* in each case.

In measuring for moments the distances must be taken on a line passing through the fulcrum and at right angles to the direction of the force. In the case of the

FIG. 17

lever safety valve the holding-down force is gravity, which acts vertically. A line at right angles to the vertical is horizontal, so that distances should be measured in a horizontal direction as through *A B C*, Fig. 20, and not on the lines *x x* or *y y*.

In determining the distance *a c*, Figs. 18 and 19, do not get bothered about the piece of lever which extends back of the fulcrum. I have heard interminable arguments upon the subject, but it takes care of itself. The more metal there is back of this point the nearer the center of gravity is to the fulcrum. If there were as much weight to the left of the pin in Fig. 17 as to the right, the center of gravity would be at the pin; the lever would balance over the pin as it did over the knife edge and not bear on the stem at all.

To apply this, suppose that the lever of a 3-inch valve weighed six pounds, that the distance *a c*, Fig. 18, between the fulcrum and the center of gravity was found to be 15 inches, and the distance *a b* from the fulcrum to the point at which the pin bears 2¼ inches. The moment of the lever must be:

$$6 \times 15 = 90 \textit{ inch-pounds.}$$

The moment of the lifting force must equal this, and that moment is 2¼ times the force. Then the force must be:

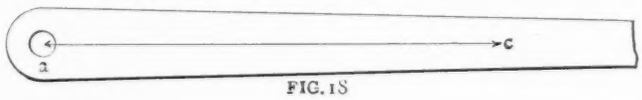


FIG. 18

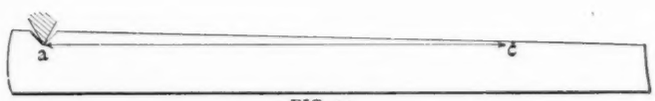


FIG. 19

$$90 \div 2\tfrac{1}{4} = 40 \textit{ pounds}$$

for

$$2\tfrac{1}{4} \times 40 = 90 \quad \textit{and} \quad 15 \times 6 = 90.$$

Since a force of 40 pounds is to be exerted upon 7.0686 square inches, the force per square inch would be:

$$40 \div 7.068 = 5.66 \textit{ pounds.}$$

The combined effect of the valve and stem and of the lever of the 3-inch valve in question would be:

$$0.212 + 5.66 = 5.87 \textit{ pounds.}$$

Columns 5 and 6 o the table already referred to give the weights of levers and the distances of their centers of

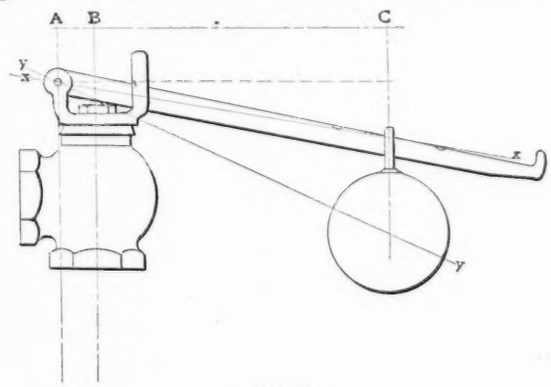


FIG. 20

gravity from the fulcrum as ordinarily found, and column 7 gives the pressure per square inch on the valve necessary to lift such levers. Column 8 gives the sum of the respective values in columns 4 and 7, i.e., the pressure per square inch required to lift the valve and stem and the lever. It will be seen that the values run fairly even for all sizes of valves, and that by using seven or eight pounds as an allowance as in the above examples, results can be attained with the simple rules which will be within a pound or two of right.

Complete Safety-valve Rules

It is evident that any complete rule for the safety valve must include the separate treatment of the valve and stem, the lever and the ball as factors in holding the valve to its seat.

Moment of the lifting force = moment of ball *plus* moment of lever *plus* moment of valve and stem.

The lifting force consists of the pressure per square inch into the area of the valve, and its moment is the product of the force by its distance from the fulcrum. Expanded then the above becomes:

Pressure *times* Area *times* Distance of stem from fulcrum = Weight of ball × distance of its center of gravity from the fulcrum *plus* weight of lever × distance of its center of gravity from fulcrum *plus* weight of valve and stem × distance of their center of gravity from the fulcrum.

In order to find one of these quantities we must know all the rest, and consequently since the missing quantity can be but on one side of the equal mark we can figure the combined value of the quantities on one side of the equation (that is, in one set of brackets). Then we can work out the operation indicated on the other side as far as we can go. If the missing quantity is on the left-hand side of the equation it can be found by dividing the value of the other side of the equation by the product of the two known factors on the left-hand side.

TO FIND THE PRESSURE AT WHICH A CERTAIN VALVE WILL BLOW OFF:

Multiply the weight of the ball, of the valve and stem and of the lever, each by the distance of its center of gravity from the fulcrum and add the products. Multiply the area of the valve by the distance of its center from the fulcrum and divide the sum above found by the product. The quotient will be the pressure required.

Or more briefly:

Divide the sum of the moments of the valve, lever and ball by the product of the area of the valve and distance from the fulcrum.

EXAMPLE—At what pressure will a 3-inch valve blow off with the stem 2¼ inches from the fulcrum, valve and stem weighing 1½ pounds, lever weighing 6 pounds, having its center of gravity 15 inches from the fulcrum and weighted with a 48-pound ball 24 inches from the fulcrum?

Pressure × Area = 7.0686 × Distance 2¼ = Product = 1590.435

48 × 24 = 1152
6 × 15 = 90
1.5 × 2½ = 3.375
Sum of moments = 1245.375

1245.375 ÷ 1590.435 = 78.3 *pounds.*

This is all that we shall be likely to wish to find on this side of the equation, for the distance of stem is fixed and the area determined by other considerations.

The other two things that interest us are the weight of the ball and its distance from the fulcrum.

TO FIND WEIGHT OF BALL OR ITS DISTANCE FROM FULCRUM:

Multiply the pressure by the area and by the distance of the stem from the fulcrum. The product is the moment of the force.

Multiply the weight of the valve by the distance of the stem from the fulcrum; multiply the weight of the lever by the distance of its center of gravity from the fulcrum, and add the products.

Subtract the sum of the products just found from the moment of the force, and the difference is the moment of the ball.

Divide the moment of the ball by the weight of the ball and the quotient is its distance from the fulcrum.

Divide the moment of the ball by the distance from the fulcrum and the quotient is the weight of ball required.

EXAMPLE—What weight of ball at the same distance would be required to allow the valve given in the previous example to blow at 75 pounds, and at what distance would the 48-pound ball there given have to be placed from the fulcrum to produce the same result?

Moment of force: 75 × 7.0686 × 2¼ = 1192.826; 93.375; 1099.451 moment of ball.

Weight × distance of ball *plus* 6 × 15 = 90.000 *plus* 1.5 × 2.25 = 3.375; 93.375 sum of moments, valve and lever.

$\frac{1099.451}{48}$ = *inches*, distance of ball.

$\frac{1099.451}{24}$ = *pounds*, weight of ball.

SPRING-LOADED OR POP SAFETY VALVES

I am sometimes asked for a rule for calculating the pressure at which a spring-loaded valve will blow off. I do not know of any which is reliable that does not involve the determining by experiment of the force required to compress the spring, and if you are going to do this you may as well determine by experiment at what pressure the valve will blow off. In practice nobody thinks of computing the spring-loaded valve. If they want it to blow off at 120 pounds they procure a suitable spring

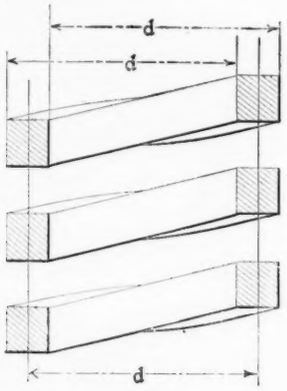


FIG. 21

from the makers and turn down upon the binding nut until the valve will blow experimentally at the desired pressure. The pressure at which a spring will yield depends not only upon the shape and size of the material of which it is made, the diameter, number, and pitch of the coils, all of which are measurable and determinable, but upon the nature and condition of the material itself. You can readily appreciate that a spring of brass would compress with less pressure than one of steel, similar in every other respect, and that there is such a wide difference in steels that there will be a great deal of difference in the action of steel springs according to the kind of metal, degree of temper, etc. The best rule of which I know is the following:

TO FIND AT WHAT PRESSURE A VALVE WILL LIFT WITH A SPRING OF GIVEN DIMENSIONS AND COMPRESSION:

Multiply the compression in inches by the fourth power of the thickness of the steel in sixteenths of an inch, and by 22 for round or 30 for square steel. Product I.

Multiply the cube of the diameter of the spring, measured from center to center of the coil (as on the line d, in Fig. 21) in inches, by the number of free coils in the spring, and by the area of the valve in square inches. Product II.

Divide Product I by Product II and the quotient will be the pressure per square inch at which the valve will blow off.

I suppose the weight of the valve and of the spring should in strictness be added to Product I, when the construction is such that the valve supports the spring; but inasmuch as the values 22 and 30 are guessed at it will not pay to go into refinements in other directions. I never compared the result of this rule with an actual valve and have no notion as to how nearly it would come to the truth. It is based on a formula adopted by a committee of Scotch engineers and shipbuilders. I have corresponded with the manufacturers of pop safety valves as to the accuracy of the formula, and they say they can proportion and calibrate their springs only by experience and experiment. However, I give you the rule as best I know, and for what it is worth. If you have a spring-loaded valve calculate it by this rule and see how nearly it comes to the point at which the valve will blow off. I should be interested to know the result of your experiment.

With a dead weight or a lever-loaded valve the force required to lift it remains the same, no matter how high the valve lifts. The weights weigh no more if they are raised an inch or two, and the leverage does not change, but with the spring-loaded valve the more the valve lifts

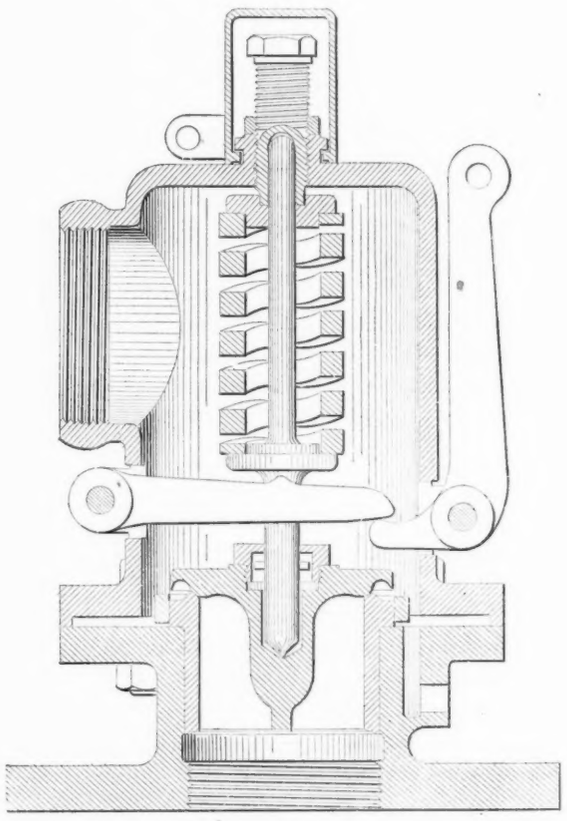


FIG. 22

the more the spring is compressed, and the more force is required to compress or hold it. It follows then that if an ordinary valve were loaded with a spring it would simply crack open and commence to sizzle when the pressure equaled the force at which the spring was set, and that if this were not enough to relieve the boiler the pressure would have to increase, opening the valve more and more until the steam blew off as fast as it was made.

But the ideal valve should stay on its seat until the pressure reaches the desired limit, then open wide and discharge the excess. This result is accomplished by the construction shown in Fig. 22. With the first opening of the valve the steam passes into the little "huddling chamber" made by the cavity near the overhanging edge of the valve and a similar cavity surrounding the seat. The pressure which accumulates here, acting on the additional area of the valve, raises it sharply with the "pop" which gives the valve its name, and it is sustained by the impact and reaction of the issuing steam until the pressure has subsided sufficiently to allow the spring to overcome these actions.

The outside edge of the lower trough in the valve shown is composed of an adjustable ring which may be screwed up or down so as to diminish or increase the distance between the overhanging lip of the valve and its own inner edge, controlling the outlet from the chamber; and the diminution of pressure or the "blow back" required to allow the valve to seat so that the valve opens wide at a given pressure and seats promptly without sizzling or chattering when the pressure has been reduced a certain amount depending upon the adjustment of the ring. The various makers have adopted different devices for adjusting the ring or other device for controlling the outflow from the huddling chamber.

THE CAPACITY OF SAFETY VALVES

Let us next consider the capacity of valves; how large a valve is required for a given boiler. Most of the rules deal with grate surface and the area of the valve; the rule adopted by the U. S. Board of Supervising Inspectors being one square inch of valve area for each two feet of grate area. That the valve should be proportioned to the grate surface seems proper because it is the grate surface, and not the heating surface, which determines and limits the capacity of a boiler. To a given grate surface, however, we should apportion a sufficient amount of area of opening, and this area of opening is not proportional to the area of the valve but to the diameter and lift. A valve one inch in diameter has an area of 0.7854 of a square inch, but that does not mean that there will be an opening of 0.7854 of a square inch for the steam to escape. If the valve is flat, as in Fig. 23, the area opened for the discharge of steam will be the circumference of the valve multiplied by the lift. The circumference is

$$Diameter \times 3.1416 \qquad (1)$$

and the area of the complete circle is

$$Diameter \times 3.1416 \times \frac{Diameter}{4} \qquad (2)$$

and the area for the escape of steam is

$$Diameter \times 3.1416 \times Lift. \qquad (3)$$

When the lift is one-quarter the diameter, or

$$\frac{Diameter}{4},$$

the area for the escape of steam is the same as the area of the circle; formula 3 is the same as formula 2.

When a flat valve has lifted a quarter of its diameter it has reached the limit of its capacity to discharge steam. It doesn't do any good to lift higher, for the area around the edge of the valve is already as large as the area of the valve itself and the capacity of the valve is proportional to the area or the square of the diameter. In practice, however, the lift of valves is much less than one-quarter of their diameter, and for a given lift the area for the escape of steam is proportional to the circumference or the diameter rather than to the area. Most of the rules, however, as above stated allow a given amount of valve area to a square foot of grate surface, and make the allowance liberal enough to include all conditions. For instance, the rule of the U. S. Board of Supervising Inspectors calls for one-half a square inch of valve area for each square foot of grate surface. A 4-inch valve has about 12 square inches of area and would thus take care of 24 square feet of grate. It would not be possible to burn over 25 pounds of coal per square foot of grate per hour with natural draft, nor to evaporate over 12 pounds of water with a pound of coal, so that the boiler could not possibly make more than

$$25 \times 12 \times 24 = 7200 \text{ pounds of steam per hour,}$$

or

$$7200 \div (60 \times 60) = 2 \text{ pounds of steam per second.}$$

Now the weight of the steam which will escape through a given aperture per second is given by the following formula:

$$Wt. = \frac{Pressure \times Area}{70}$$

that is, the weight in pounds which will escape in a second is equal to the absolute pressure in pounds per square inch multiplied by the area in square inches and divided by 70.

On the other hand the area required to discharge a given weight is

$$Area = \frac{Weight \times 70}{Pressure}$$

that is, the weight in pounds to be discharged per second multiplied by 70, and divided by the absolute pressure equals the required area. Now we have found that with a rate of combustion practically impossible, with natural draft, and a practically unattainable evaporation per pound of coal, the most steam that the boiler with 24 square feet of grate surface could furnish is 2 pounds per second. The area required to discharge this at 70 pounds pressure, absolute, is

$$\frac{2 \times 70}{70} = 2 \text{ square inches.}$$

The 4-inch valve which this boiler would require would have a circumference of practically 12 inches, and would need to lift only one-sixth of an inch to furnish the two square inches of opening necessary to discharge the steam, for

$$12 \times \tfrac{1}{6} = 2.$$

One-sixth of an inch is only one-twenty-fourth of the diameter of the valve. You see that this simple rule gives an ample margin, requiring but a small lift to discharge more steam than the boiler can possibly make. It is altogether useless and nonsensical to figure the areas of opening to four places of decimals involving with beveled seats complicated operations with sines and cosines, in a calculation which involves no accuracy but which requires simply a result which shall be amply large to cover any emergency likely to be encountered in practice. It is like trying to measure the distance to the next town in feet and inches, in order to answer a man who would be abundantly satisfied to know that it was about three-quarters of a mile. You may be sure that a valve which has a square inch of area for each two square feet of grate surface will liberate all the steam that can be made by the coal that you can burn on that grate surface, so long as the valve is free and in good condition. It is quite probable that a smaller valve would do, but in a matter of this kind we want to provide not the smallest that will possibly do but enough capacity to be absolutely safe. For all purposes of ordinary practice, therefore, divide the grate surface by 2, which will give you the valve area required and you can find the corresponding diameter by multiplying the square root of the area by 1.128. Don't carry your decimals out too far because you will have to take the nearest commercial size after all.

Here is a rule w ich will give you the diameter of the valve in inches at once:

Multiply the square root of the grate surface by 0.8.

This would be particularly handy when the grate is square, or nearly so, for then the length would be the square root of the area.

You can see how the rule is made, or rather, makes itself.

By the supervising inspector's rule the valve area required equals the grate surface divided by 2.

$$area = \frac{grate\ surface}{2}.$$

The diameter is the square root of the quotient of the area divided by 0.7854.

$$diameter = \sqrt{\frac{area}{0.7854}}.$$

And since in this case the area equals one-half the grate surface the diameter will be the square root of one-half the grate surface divided by 0.7854.

$$diameter = \sqrt{\frac{grate\ surface}{2 \times 0.7854}};$$

or,

$$diameter = \sqrt{\frac{grate\ surface}{1.5708}}.$$

We can get rid of the square root in the denominator by finding it once for all. It is 1.25 very nearly. So our formula becomes

$$diameter = \frac{\sqrt{grate\ surface}}{1.25}.$$

Dividing by 1.25 is just the same as multiplying by $\frac{1}{1.25}$, and as $\frac{1}{1.25} = 0.8$, the multiplication is easier, so we have

$$diameter = \sqrt{grate\ surface} \times 0.8.$$

The grate surface will never be so large that the square root cannot be easily determined with sufficient accuracy mentally. If it is between 25 and 36 the root is between 5 and 6. The square of 7 is 49, of 8, 64, etc., so that by trial the root can be determined approximately. Here is an easy trick to get the square of a number with two figures ending in 5:

Multiply 1 plus the left-hand figure by the left-hand figure, and annex 25 to the product.

What is the square of 35?

The left-hand figure is 3. Three plus 1 is 4, and $4 \times 3 = 12$. Annex 25 and get 1225.

This rule works just the same when the 5 is a decimal, only in that case the annexed 25 is a decimal, too, and will enable you to determine instantly by inspection the nearest number advancing by halves to the square root. As the sizes of safety valves advance by half inches, the nearest root determined in this way will be sufficiently accurate, as we have to take the nearest commercial size anyhow.

What is the square of 6 5?

Six plus 1 = 7; $7 \times 6 = 42$; add 25, which in this case will be a decimal fraction, there being two places to point off, and get 42.25.

In this way you can square 1.5, 2.5, 3.5, etc., and this is as near as it is ever necessary to get a root in the above formula. Suppose, for instance, you had 58 square feet of grate surface. What is the square root? Seven times 7 = 49, and $8 \times 8 = 64$. It must be between 7 and 8; $7.5 \times 7.5 = 56.25$.

That is near enough to 58. The square root of 58 is really 7.615. Multiplying this by 0.8 we get $7.615 \times 0.8 = 6.092$, which is practically a 6-inch valve. We should have got at the same result if we had taken the square root as 7.5, for $7.5 \times 0.8 = 6$.

When the grate surface is over 30 or 40 feet it is better to get the required capacity by putting on two valves than by using one large one. In fact it is a pretty good plan to have two safety valves anyway. There is a great deal of responsibility on that little appliance, and many of the most destructive of boiler explosions would have been avoided by an operative safety valve of sufficient capacity. So many little things can occur to make it hold against a destructive pressure, even when the attendant follows the usual directions to raise it from its seat daily, that prudence dictates the use of an auxiliary valve. It would be a remarkable coincidence if both stuck at the same time without criminal negligence.

The amount of opening of an ordinary lever safety valve is determined by the amount of surplus steam to be delivered. If the boiler is making more steam than is to be taken out of it the pressure will increase, and when

it reaches an amount sufficient to overcome the weight of the ball, etc., the valve will be raised a little from its seat and the steam will escape. If the opening thus afforded is sufficient with the other drafts on the boiler (such as the supply to the engine, etc.) to allow all the steam the boiler is making to escape the valve will not open any wider, but if not the pressure will continue to increase and force the valve open until the steam can escape as fast as it is made. As the surplus production of steam decreases, as by closing the dampers or a greater demand by the engine, the valve gradually settles down to its seat again.

On account of its greater lift and effective discharging area the pop valve is allowed by the Board of Supervising Inspectors three square feet of grate surface per inch of area instead of two, as with the ordinary lever valve.

We have seen that the escape of steam through an opening of given size is proportional to the absolute pressure. Twice as much steam will go out of an inch hole in a minute with 190 pounds behind it as with 95 pounds. It is, I presume, for this reason that the inspectors only require a square inch of valve area for every 6 feet of grate surface on boilers carrying a steam pressure exceeding 175 pounds gage.

I have said that although the area effective for the escape of steam is not proportional to the area due to the diameter of the valve, and although the latter area is that used in the formula for capacity, the allowance is so liberal that it is practically useless to figure the former. It may be interesting, however, to know how to figure it, and a treatise on the safety valve would hardly be complete without directions for so doing.

With a flat valve we have already seen that the area for the escape of steam is the lift of the valve multiplied by its circumference. With a bevel-seated valve in which the valve does not lift out of the seat the area AA, Fig. 24, is that of a frustum of a cone, Fig. 25. Now to find this area the rule is to add the circumference of the greater circle to the circumference of the lesser CD; divide by 2, and multiply by the slant hight CA. In other words, to multiply the average length of the strip which

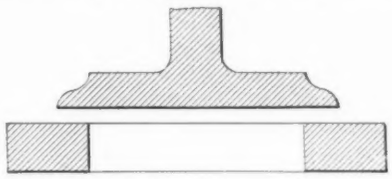


FIG. 23

would be made by flattening this surface out by the width of that strip. To work this rule out would take us too far into trigonometry, but I will give you the rule:

(1) *Multiply the diameter of the valve by the lift, by the sine of the angle of inclination and by 3.1416.*

FIG. 24

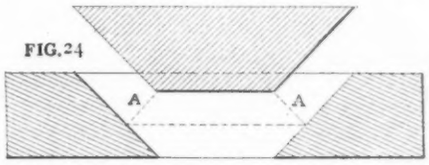
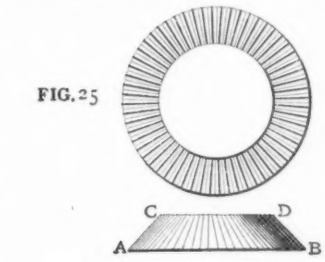


FIG. 25



(2) *Multiply the square of the lift by the square of the sine of the angle of inclination, by the cosine of this angle and by 3.1416.*

(3) *Add these two products.*

The U. S. rules require a bevel of 45 degrees, and most valves are made with seats of that degree of inclination. For such a valve the rule becomes:

(1) *Multiply the diameter of the valve by the lift and by 2.22.*

(2) *Multiply the square of the lift by 1.11.*

(3) *Add these two products.*

When a valve with a beveled seat lifts clear of the seat as a valve with a slight bevel may, the area of the opening is computed by the above rule for a lift which would raise it to the upper level of the seat, and to this is added the circumference of the valve multiplied by the lift above the seat level.

1	2	3		4	5	6	7	8	9	10	
Diam'r of Valve.	Area of Valve.	Weight of Valve and Stem.		Pressure Required to Lift Valve and Stem.	Weight of Lever.	Distance of Center of Gravity from Fulcrum.	Pressure Required to Raise Lever.	Pressure Required to Raise Valve, Stem and Lever.	Length of Lever.	Distance of Stem from Fulcrum.	Weight of Ball Ordinarily Furnished.
In.	Sq. In.	Pounds.		Pounds per Sq. In.	Pounds.	Inches.	Pounds per Sq. In.	Pounds per Sq. In.	Inches.	Inches.	Pounds.
⅜	0.1104	0.125		0.113	0.125	3.25	5.89	6.003	6.31	0.69	1.56
½	0.1963	0.156		0.7947	0.140	3.0	2.85	3.6447	5.62	0.75	3.2
⅝	0.4418	0.187	0.14	0.423	0.343	4.812	4.98	5.403	9.50	0.75	5.5
¾	0.7854	0.187	0.23	0.238	0.521	7.75	8.92	8.538	14.87	1.0	8.12
1	1.2272	0.312	0.34	0.254	0.432	7.312	5.65	5.904	14.4	1.19	9.62
1¼	1.7671	0.437	0.60	0.247	0.488	6.875	2.87	3.117	13.4	1.19	15.37
1½	3.1416	0.542	0.75	0.172	0.424	13.312	7.37	7.542	26.1	1.44	19.0
2	4.9087	0.8395	0.97	0.171	0.875	15.25	7.24	7.584	30.0	1.87	29.0
2½	7.0686	1.339	1.5625	0.189	2.5	18.625	6.67	6.859	37.12	2.25	38.0
3	9.6211	1.8	2.33	0.187	3.5	21.75	6.52	6.967	43.1	2.25	48.5
3½	12.5666	2.371	2.75	0.189	4.75	19.0	5.78	6.008	38.375	2.25	68
4	15.9043	3.0	3.50	0.189	6.0	22.0	4.78	4.969	44.5	2.37	88
4½	19.6350	4.125	4.75	0.210	10.50	23.25	8.75	9.077	46.5	2.75	110
5	28.2744	5.87	6.45	0.324	10.75	25.25	6.33	6.75	53.5	2.5	140
6			8.62	0.208	15.0	21.25	32.0	29.62	5.53	7.22	10.62	12.05	6.36	5.768	12.355	6.78	54.5	59.50	74.5	2.625	3.50	3.5	139.0	220	139.0

POWER

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The Defective Tube Scandal

What has become of the inquiry into the charges made by Frank L. Emmett that the Shelby Steel Tube Company worked off a lot of condemned tubes on the United States Government? If it is a fact it ought to be established, not only on general principles, but because those tubes are somewhere where something more than human life may depend upon them. If it is not so, the Shelby company should be relieved of the stigma. The matter ought not to be allowed to die, nor to be pigeon-holed.

Lap-seam Boilers

Joseph H. McNeill, chief inspector of boilers for Massachusetts, has rendered a report upon the Lynn explosion, described on pages 84, *et seq.*, of our February issue. It reveals nothing of interest which our article did not contain, but concludes with the following sensible remarks upon the continued use of boilers of this type:

"The further installation in this Commonwealth of steam boilers with shells or drums over 40 inches in diameter, constructed with longitudinal lap joints, should be prohibited by law, provision being made for the completion of boilers of this type in process of construction or contracted for.

"The increase in cost of a steam boiler with longitudinal joints of the butt-strap form is very small, compared with that of one with lap joints.

"There is no case recorded of the explosion of a steam boiler through the failure of a butt-strap joint."

The Steam Engineer and the Gas Engine

As we have pointed out before, a gas engine of any considerable output is somewhat more difficult to keep in proper condition than any standard type of steam engine. It is needless to go into details here; the purpose of the present writing is not to argue the question of relative complexity, but to point out that the increased attention demanded by the gas engine should be no cause for discouragement on the part of the operating engineer. It is undeniably true that the best-informed steam engineer could not take charge of a plant of double-acting gas engines and keep them going properly from the start, except through phenomenal good luck, unless he had had previous training in that class of work. It is also just as true, however, that a man who is inherently capable of becoming a good steam engineer is also inherently capable of becoming a first-class gas engineer; in other words, the differences between the two classes of work are not such as to require any higher degree of mentality in the gas engineer than in the steam engineer. Probably a more careful type of man is required, but it would be rather academic to differentiate in this respect; if a man is as careful by nature as he should be to have charge of an important steam plant, that is sufficient for handling gas engines, too. Such a man will very soon learn what degree of alertness a gas-engine plant demands of him. But your happy-go-lucky steam engineer, who allows an engine to pound until he accumulates a sufficient store of energy to locate and remedy the trouble, has no place in a gas-engine plant. Of course, such a man really has no logical place in a steam plant, either, but while he might be able to pull through there he would inevitably come to grief in a gas-power plant.

The amount of preliminary training that an earnest steam engineer requires to fit him for coping with gas engines is surprisingly small—not nearly so extensive as is, or should be, the extent of preparation requisite to enable a steam engineer to deal with electrical machinery. Therefore, we assert, there is no reason for the experienced and conscientious steam engineer to feel discouraged or apprehensive on the score of an impending change in his power equipment from steam to gas.

One of the builders of large gas engines has adopted the very sensible policy of giving the engineer in charge a preliminary training in the care of the engines whenever they put in a plant. A man experienced in the actual operation of gas engines is left with the plant when it is started up, and he remains there until the regular operating engineer has either become proficient or demonstrated a hopeless inability to do so; thus far, no men of the "hopeless" type have been encountered—which goes to bear out our argument that a good steam engineer will make a good gas engineer, or any other kind.

Comparative Test of Alcohol, Kerosene and Gasolene as Fuel for Engines

An interesting test was made recently to determine the relative value of gasolene, kerosene and denatured alcohol as fuel for automobile engines. The test was not absolutely conclusive, of course; no single test of this character ever is. But the results seem to show that alcohol has a considerably higher fuel value than its thermal value would indicate. The average calorific values of gasolene, kerosene and denatured alcohol, respectively, are 18,000, 19,000 and 11,000 B.t.u. per pound, or 110,000, 126,000 and 73,000 B.t.u. per gallon at ordinary temperatures. The trial under consideration showed fuel consumptions of 0.0876 gallon per ton-mile with gasolene, 0.1076 gallon with kerosene, and 0.119 gallon per ton-mile with denatured

alcohol, which figures are obviously out of all proportion with the relative calorific values of the fuels.

The test consisted of driving three Maxwell automobiles of exactly similar design, excepting the fuel-handling apparatus, from New York to Boston, the actual distance traversed being 249 miles. The weights of the cars and their fuel records are given below:

Fuel.	Car Weight.	Gallons Used.	Ton-Miles.	Gallons per Ton-Mile.	B.t.u. per Ton-Mile.
Gasolene.....	2270	24.75	282.6	0.08758	9,636
Kerosene.....	2520	33.75	313.74	0.10757	13,554
Alcohol.....	2750	40.75	342.4	0.11902	8,688

The differences in the car weights are due chiefly to the differences in the weights of the passengers, the figures given in the table being the complete weights of the cars, passengers and equipment.

With gasolene at 18 cents per gallon, denatured alcohol would have to sell at 13¼ cents per gallon in order to give the same results per unit of cost, according to the figures in the table; to be on the same plane with 20-cent gasolene, alcohol would have to be sold for 14.7 cents.

It will be noticed that the alcohol engine shows a much better thermal efficiency than either of the others, on the basis of ton-miles. The principal reason for this, doubtless, is that higher compression was employed in the alcohol-engine cylinder.

The poor showing of kerosene as compared with gasolene was probably due to the difficulty of vaporizing it, resulting in very incomplete combustion. The observer's report, however, says nothing on this point. Even with the imperfect performance that the record of the kerosene-driven car makes obvious, kerosene at 14.7 cents is on a par with gasolene at 18, and the price would have to exceed 16¼ cents in order to put kerosene at a disadvantage with 20-cent gasolene. With better means of utilization than the ordinary mixture-compressing engine, kerosene should make a much better showing.

The Barbarities of Peace

A movement to reduce the number of fatalities and injuries in the industrial army as well as in accidents upon transportation lines was inaugurated in an impressive manner by a banquet at the Waldorf-Astoria in New York on the evening of January 28. Dr. Josiah Strong, president of the American Institute of Social Service, under whose auspices the movement is started, presided and placed the importance of the subject before his hearers by the startling announcement that the number of persons killed and injured in peaceful pursuits in this country is greater than the collective casualties of a great and continuous war; that the total number killed in the civil war was less than one-half the number now being killed in an equal length of time by accidents, for the most part preventable.

It is a matter of common knowledge that we are far behind European countries in this respect. Not only are their injuries in proportion to the number of passengers carried upon their railroads very much less than ours, but their statistics of boiler explosions and other industrial accidents reveal by comparison a condition in America which calls for active and immediate attention. The call is urgent enough from the humanitarian point of view, but as Carroll D. Wright pointed out it has an important commercial and financial bearing. It was said that the number of people killed or injured in the various industries of the United States amounts in a single year to 500,000. Take the usual damage-court estimate of the value of a human life at five thousand dollars, and damage for injuries usually exceed those for death, and where do you come out? And who pays it? Not the employer. He may pass over the money but his business must show a net profit, and this disbursement goes into the cost of his product or service and is paid for, ultimately, by the consumer.

The entire public then is interested in the success of this movement in several ways. It may be your turn next, to be ground up in a railroad wreck, to be dropped in an elevator, to be hit by a trolley car, to be electrocuted by a live wire, to be hurled to death by a boiler exploding under the sidewalk upon which you are walking or under the building in which you work, or to be crushed in the wreck occasioned by a mass of iron projected from a bursting fly-wheel blocks away. These things may happen to anyone. If you are a workman in a factory where there is moving machinery or perilous processes the risk is the greater. If you escape them all you are paying your proportion of the cost of those which do occur in the consequently increased cost of the necessities of life, and behind all this are the physical suffering and heartaches and bereavements and privations of those who are caught or whose breadwinners go down in the industrial war.

The principal speaker of the evening was Governor Hughes and one of the telling points of his most forcible speech was his laying bare of the inconsistency of the common-law ruling whereby the employer escapes liability for injuries by proving them to have been caused by the negligence of a fellow employee. The "freedom of contract" which is so frequently urged is a farce, for the workman is generally forced to accept employment upon the employer's own terms or starve.

The bearing of this movement upon the field in which we are directly interested comes in the encouragement of means for the prevention of boiler explosions and fly-wheel accidents; for the immediate stoppage of shafting and machinery when a person becomes involved; for the betterment of conditions in underground engine- and boiler-rooms, where the temperature is unbearable under normal conditions, and escape in case of a bursting pipe or boiler tube practically impossible. Foreign governments not only provide for inspection of boilers but require them to be set so that in case of a rupture the escape of the steam, etc., would be into an open court instead of into the fire-room. We have consistently advocated the examination and licensing of the men in charge of steam plant, not as students of thermodynamics and the higher mathematics, but as good, safe, reliable, practical men who know whether their plant is safe or not and how to operate it without danger to the surroundings. With this kind of a man in charge, with boilers properly designed and built, and engines provided with safety devices against destructive overspeed and piping properly laid out and supported, the casualties in power plants could be reduced to a small percentage of the present record.

President Roosevelt in a letter read at the dinner expressed his warm approval of the movement and the large and representative attendance, including such men as Baron des Planches, ambassador to the United States from Italy, M. Ebray, consul-general from France, Hon. Seth Low, Count Massigilia, from Italy, Chas. Stewart Smith, ex-president of the New York Chamber of Commerce, *et al.*, was an earnest of the impulse which it has already received. An exhibition of safety devices described on page 185, under the direction of Dr. William H. Tolman, is one of the methods adopted of eliciting public interest in the matter.

Regarding Graft in the New York Post Office

Charles T. Porter calls attention to the fact that his engine was put into the New York post-office in 1881, as is evident to all readers of the entire series of his articles, and that the letter of Alfred Brooks Fry, in the February issue, serves no other purpose than to exonerate the present administration from complicity in the graft proceedings recounted in his "Reminiscences."

"Unavoidably Postponed."

Several articles of unusual interest which were announced a few weeks ago as being intended for publication in this issue have been unavoidably delayed. These will all appear next month.

NEW THINGS FOR THE POWER PLANT

The Hoppes Class R Feed-water Heater

Fig. 1 herewith is a side sectional elevation of a Hoppes class R open feed-water heater, Fig. 2 being an end sectional view of the same. Although this heater has been on the market several years, it has recently been improved and embodies features not heretofore shown. The shell of the heater is cylindrical and the heads are "bumped," a design calculated to resist pressure, and also to prevent pulsations due to the impulses of the exhaust. The interior of the heater is provided with layers of trough-shaped pans arranged in tiers and designed to afford a large amount of heating surface. To avoid corrosion the pans are of cast-iron, as are also the bottom of the shell, the lower ends of the center posts and all other parts with which the water comes in contact. The shell may be entirely cast-iron, however, if desired. In the back end of the heater is located a large oil catcher, through which the exhaust passes. See Fig. 1.

The principle of operation is to provide that the flow of water from the cast-iron troughs at the top be so gradual that the water will be distributed over the edges of the pans in a thin film, and flow over the sides and ends so gently that it will follow the bottom contour of the water will, when liberated by the heat, form mostly on the under side of the pans and hence will not detract from the efficiency of the process, as the same direct action of the exhaust on the water will continue as when the pans are clean.

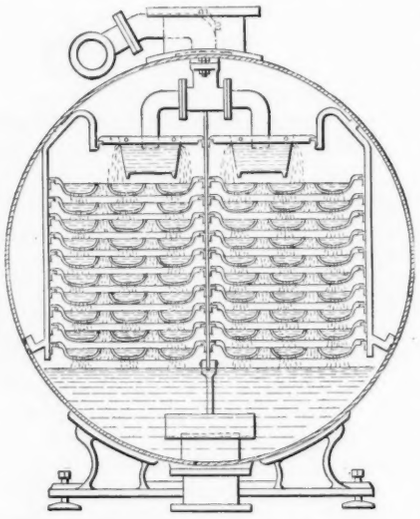


FIG. 2. END SECTIONAL VIEW OF HOPPES CLASS R HEATER

In open heaters of large size the regulation and distribution of water so as to obtain the best results are important. In the Hoppes apparatus the water is regulated to the heater by a float in a separate float-chamber operating a double-disk balanced valve of the company's manufacture. Branch pipes from the main feed-pipe are connected to the shell at the top and these branches have extensions inside the heater to which are attached flanged tees. To the flanges of these tees are bolted flanged inverted L-shaped pipes, the long arms of which extend below the water level of the feed troughs in the top of the heater. Disks with orifices of proper and equal size are placed between the flanges of the L-shaped feed-pipes and the tees, and by this means the water is equally distributed into the various tiers of pans.

When it is desired to feed two or more heaters in multiple, an equal distribution of water is given to all the heaters by using a single regulating valve on the main water inlet, and branching the same to the heaters to be supplied.

A feature recently added is an overflow dam, at the rear end of the heater, which is intended to act as a skimmer and also to add to the capacity of the overflow pipe by increasing the head on the outlet without causing the water to rise higher in the heater. The drip from the oil separator is piped into the chamber formed by the dam, but this may have a separate connection, if preferred. Filters are provided on request, but it is believed that the large amount of lime-catching surface obviates the necessity for their use in most cases.

These heaters are built in sizes ranging from 50 to 30,000 horse-power by the Hoppes Manufacturing Company, Springfield, Ohio.

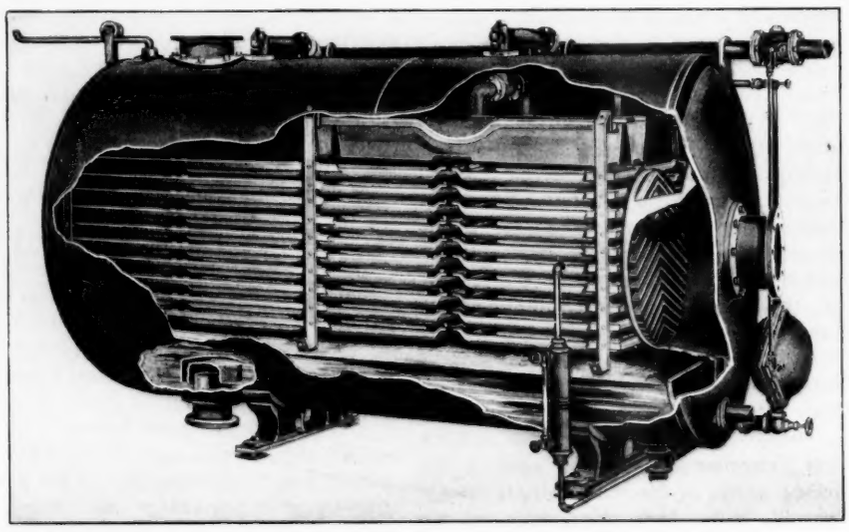
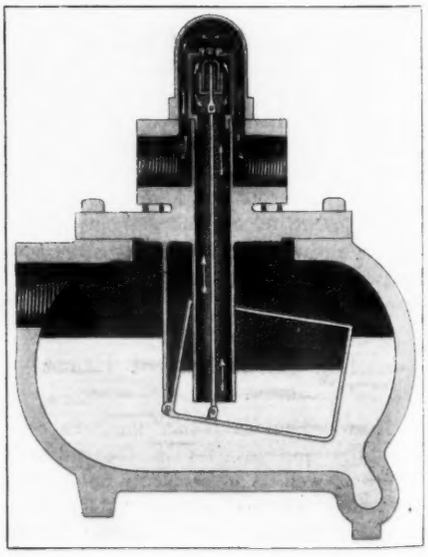


FIG. 1. SECTIONAL ELEVATION OF HOPPES CLASS R FEED-WATER HEATER

Improved Steam Trap Adapted to High-pressure Work

Attention is being directed to an improved steam trap, shown in cross-section in the accompanying engraving. Its reported performances under recent tests indicate that it will adequately handle water at exceptionally high pressures.



IMPROVED STEAM TRAP

Its action depends upon the buoyancy of a cast-iron bucket hinged on cone bearings at one side and connected to the valve

mechanism by a similarly hinged brass rod. The proportions of the bucket and the location of the valve-rod are such that at atmospheric pressure the bucket, when submerged, exerts a pull of six pounds on the valve-rod. In practice when condensation has reached the high-water line the bucket fills, sinks, and opens an auxiliary valve which is ⅛ of an inch in diameter. Pressure under the main valve is thus relieved, and it is pushed down and open instantly by the internal trap pressure upon its upper surfaces. The bucketful of water is then discharged and the bucket on rising closes both valves until the operation is repeated. No wire-drawing is possible, as the valves are always wide open or closed tightly. A feature of the trap is the location of the susceptible joints entirely above the water line.

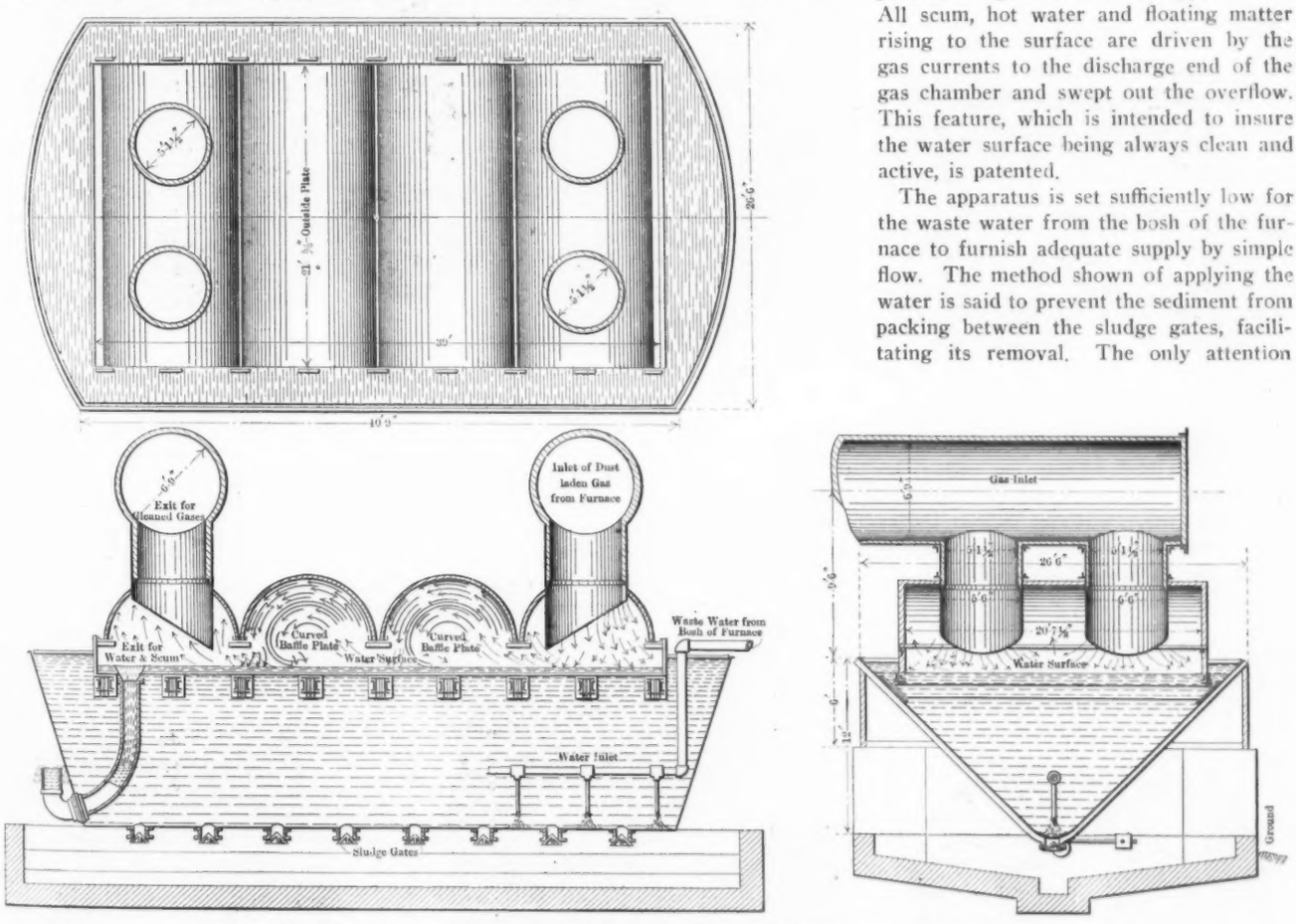
The inventors of the trap, C. E. Squires and Benjamin Silliman, of Cleveland, Ohio, made the tests referred to, with the following results: Steam alone was admitted to the trap at first until the gage indicated 600 pounds. This pressure was maintained for some time to test the special corrugated-copper gaskets used. In this test the joints remained perfectly tight. A second test was made to find the highest pressure at which the trap would operate. Enough water was pumped into the trap to cover the bucket, and, by opening the air vent, the pressure was gradually lowered until the trap began to discharge. This did not occur until the gage showed 375 pounds. Additional water was continually pumped in and discharged at this pressure for ten minutes. The action of the valve was accompanied by slight vibration. The pressure was further reduced to 300 pounds, when the valve acted smoothly, taking care of the water pumped into the trap in a continuous stream very satisfactorily.

The Pettit & Barrows Gas Washer

A new form of gas-cleaning apparatus for removing dust from blast-furnace gases has been patented by Frank Pettit and W. A. Barrows, Jr., of Sharpsville, Penn. It belongs to the class of primary cleaners, intended to treat initially large volumes of blast-furnace gases (60,000 to 75,000 cubic feet per minute), in a simple and inexpensive manner. The apparatus has no moving parts, and is designed for the cleaning of gas suitable for burning under boilers, etc.; it does not aim to perform the functions of a gas-refining plant for gas engines.

The illustration shows a gas washer suitable for use with a furnace producing 300 to 400 tons of iron per 24 hours. It consists of a water vessel with inclined sides, provided with sludge gates at the bottom for drawing off sediment, and a gas chamber, closed by a water-seal on all sides, which permits scraping the sediment from the sides of the water vessel while in operation. The gas enters at one end of the chamber, preferably by two nozzles as shown in the end elevation, for better distribution, making one impingement on the surface of the water, thence passing to the first curved baffle plate. The larger portion follows the course between the baffle and the inner side of the arched cover, causing it to be directed downward against the water surface a second time. Part of the gas is caught in front of the curved baffle, forming an eddy, and finally escaping beneath. After a repetition of this process the gas leaves at the opposite end. All scum, hot water and floating matter rising to the surface are driven by the gas currents to the discharge end of the gas chamber and swept out the overflow. This feature, which is intended to insure the water surface being always clean and active, is patented.

The apparatus is set sufficiently low for the waste water from the bosh of the furnace to furnish adequate supply by simple flow. The method shown of applying the water is said to prevent the sediment from packing between the sludge gates, facilitating its removal. The only attention required is to draw the sediment from the sludge gates every 8 or 10 hours, allowing it to flow to a settling pit, from which it is easily dredged out every 24 hours by the crane that serves the cinder pit, either to be discarded or to be formed into briquets.



PLAN, LONGITUDINAL AND CROSS SECTIONS OF PETTIT & BARROWS GAS WASHER

The top and sides of the gas chamber are brick lined to avoid undue loss of heat from the hot gas, which is usually at a temperature of 400 to 600 degrees Fahrenheit. The gases can be discharged directly from furnace into this washer, its large passages being free from clogging in case of even the heaviest slip.

A gas washer of this type has been in use for two years at the plant of the

Shenango Furnace Company, Sharpsville, Penn., during which time it is said to have demonstrated its effectiveness to the entire satisfaction of the users.

In use a thick scum gradually rises on the water surface outside the gas chamber, attaining a thickness of several inches, and dense enough to support an umbrella or walking cane.

Cook's Metallic Packing

The form of this packing used for ordinary piston-rods is shown in Fig. 1. The packing rings proper, *F* and *J*, are of babbitt metal, held to the rod by the bevel block *C*, backed up by the spring *B*. The spring guide *A*, the first piece put into the box, holds the spring centrally and prevents it from coming in contact with and cutting the rod. The bevel face of the casing *K* makes a steam-tight joint with the babbitt blocks and is concaved on the other surface to fit the convex face of the ball ring *N*, forming a ball joint which allows the packing to conform to any deflection of the rod induced by poor alinement. The plane face of the ball ring is a ground joint against the inside end of the gland *O*. A copper gasket *P* makes a steam-tight joint between the flange of the gland and the face of the stuffing-box. Packing in this form is adaptable to any stuffing-box and for use with saturated steam not to exceed 150 pounds on horizontal engines. Figs. 2 and 3 show anti-friction babbitt rings *F* and *J* upon a larger scale. The beveled face of the follower *C* fits the beveled face of the lap-joint ring *F*, which holds it firmly to the cone *J*, the former having a beveled corner *H* which fills out the remaining taper of the cone ring *J*. The bevel *H* of the ring *F* seals the opening where the cone ring is cut, thus making a steam-tight joint. The long bevel of the cone ring *J* seals the joint where the lap-joint ring *F* is cut. There is a lug, *G*, on the cone ring *J* which serves to hold the joints of the packing rings *F* and *J* in transverse section, making it impossible for steam to escape at these points. Enough clearance is cut out of the rings *F* and *J* to keep the ends from coming together, which would force the rings from the rod.

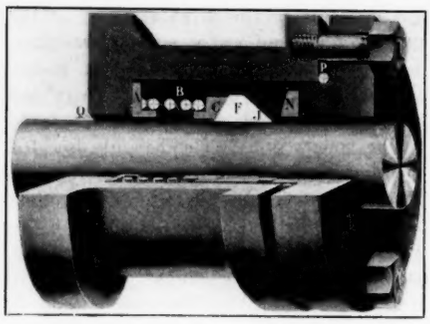


FIG. 1. COOK'S METALLIC PACKING

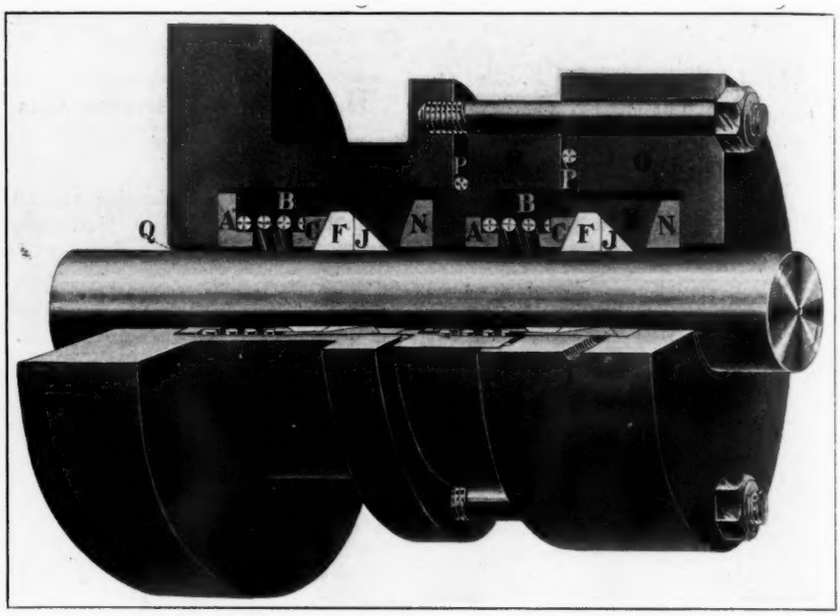


FIG. 4. DOUBLE PACKING FOR VERTICAL AND CONDENSING CYLINDERS

FIG. 2. FIG. 3.
ANTI-FRICTION BABBITT RINGS

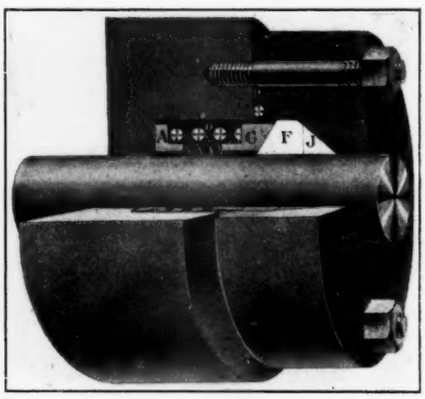


FIG. 5. SINGLE VALVE-STEM PACKING

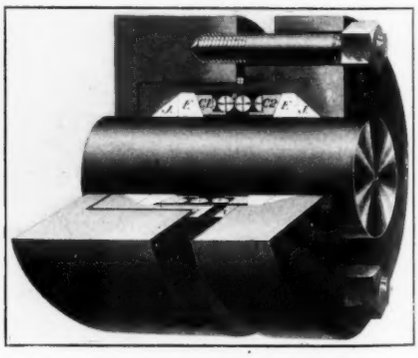


FIG. 6. DOUBLE VALVE-STEM PACKING

Fig. 4, the reference letters of which apply to the parts to which they have been assigned in the previous engravings, shows a double form of the packing used upon the rods of vertical engines and of the low-pressure cylinders of condensing engines where a high degree of vacuum is desired. The set nearest the cylinder serves to hold the steam pressure, while the outer set catches the water and conveys it off through the threaded opening *T* of the outer gland *O*. If desired, a drain pipe can be screwed into this gland and led to any convenient place. In the case of the compound condensing engines a pipe is led from the receiver to the tapped hole *T* and the receiver pressure acts to balance the atmospheric pressure, effectively sealing the vacuum. A globe valve on the pipe can be used to throttle the steam admitted and to permit at the same time the escape of any condensation. Figs. 5 and 6 illustrate the packing as applied to Corliss valve stems, the first for non-condensing engines and the other for the condensing cylinders of engines where a high degree of vacuum is to be maintained. Its departure from the piston-rod form is so slight, consisting simply of adapting the gland itself as a casing, that further description is unnecessary. It is manufactured by the C. Lee Cook Manufacturing Company, Louisville, Ky.

A New Superheater System

Hans O. Keferstein, formerly of Barberton, Ohio, and now of Brooklyn, N. Y., recently patented a new superheater system for steam boilers, which is illustrated herewith. It is designed to be located in the upper part of or superimposed on the boiler, and the novel feature of the invention consists of a series of dampers, baffle plates and partitions located between the

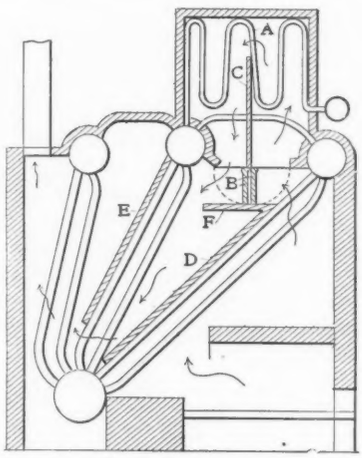


FIG. 1

superheater and the boiler flues for the more certain regulation of the gases of combustion.

Although this superheater system may be adapted to any type of boiler, for purposes of illustration the types represented in Figs. 1 and 2 have been selected.

In Fig. 1 the superheater *A* is located in a space formed above the boiler space. Swinging dampers *B* are provided between the combustion chamber and the superheater. Within the superheater partitions *C* insure the proper passage of the gases around the tubes. Baffle walls *D* and *E* and a horizontal partition *F* divide the combustion chamber into sections to guide and divert the travel of the gases.

In Fig. 2 the superheater is shown at *A*, the dampers at *B B*. A horizontal baffle plate *G*, in which is an opening to be closed by a slide *H*, is located above the furnace to deflect the gases to the rear, and a second baffle plate is shown at *I*. The swinging dampers, when closed, form a continuation of the partition *J*.

While the dampers are in the positions indicated in the sketches, the gases will flow through the furnace and the boiler passes or flues, through the superheater space and among the tubes of the superheater, thence through the boiler flues to the stack. If the dampers are partly open, part of the gas flows through the boiler passes and superheater space to the stack, and another part flows through the boiler passes to the stack. By closing the dampers the entire supply of gases may be directed through the boiler only, leaving the superheater untouched.

The dampers may be of any suitable design, as shutters, leads or single plates, and may be operated by a handle or otherwise.

Among the advantages claimed for this system by the inventor are: Accessibility for inspection and repair; easy and certain control of the temperature of the steam; avoidance of the flooding of the superheater by water; elimination of the formation of scale in the superheater due to flooding; increased durability as compared with the flooding system; reduction in cost of superheater construction; highest efficiency of fuel by the cheapest means.

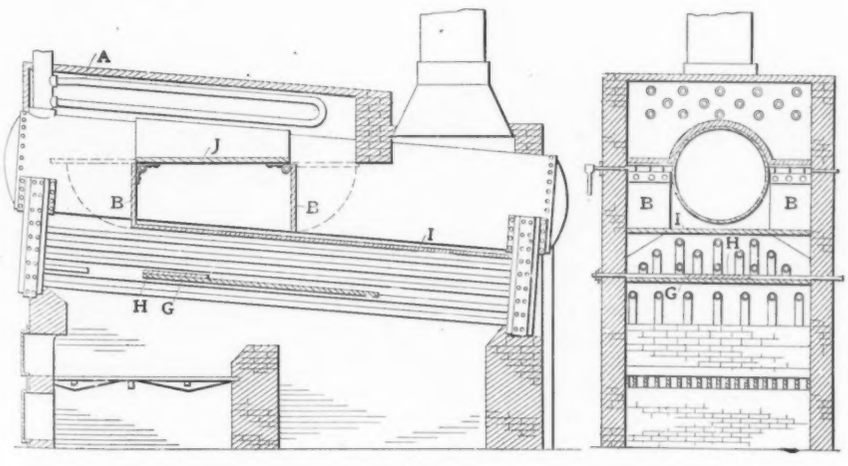
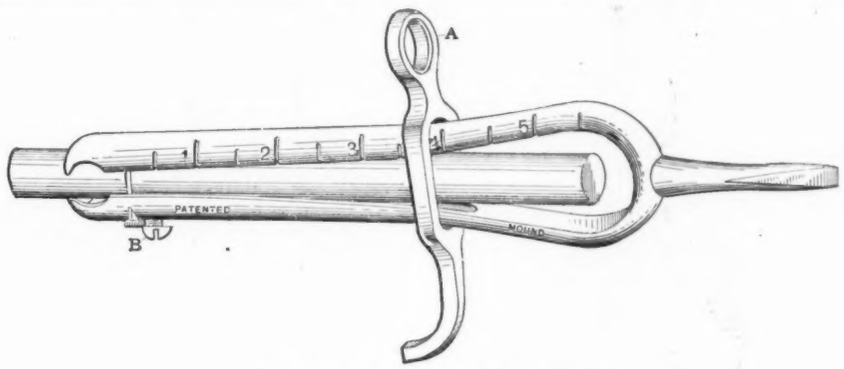


FIG. 2

The Mound Handy Arc-lamp Trimmer

Herewith is illustrated a combination tool which has been designed to not only expedite the work of trimming arc lamps, but to do it accurately, a desideratum which those who are familiar with such work will appreciate. The device comprises a carbon cutter, carbon pliers, carbon gage, carbon measure and a screw driver.

Referring to the cut, *B* is a removable cutter wheel, similar to the small wheel on a glass cutter. The carbon to be cut is inserted in the jaws of the tool until the desired length is indicated on the measure. The tongs are compressed by moving the clamp so as to hold the carbon securely, but not immovably. Then the carbon is turned around slightly, and the little cutter wheel makes its mark thereon, at which point the carbon can be readily broken, leaving even surfaces at the break.



MOUND HANDY ARC-LAMP TRIMMER

The opening *A* in the clamp is intended for gaging, as to diameter, so there can be no excuse for using carbons that are too large. The jaws of the tongs are grooved both ways like gas-pipe pliers.

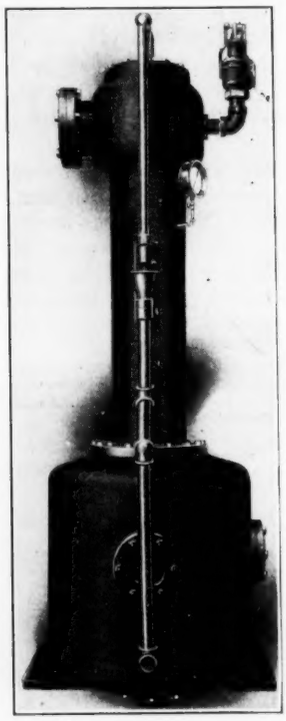
The tool is made of cast-steel heavily nickel-plated, weighs eight ounces, and can be carried in the pocket. It is the product of the Mound Tool and Scraper Company, 710-712 Howard street, St. Louis, Mo.

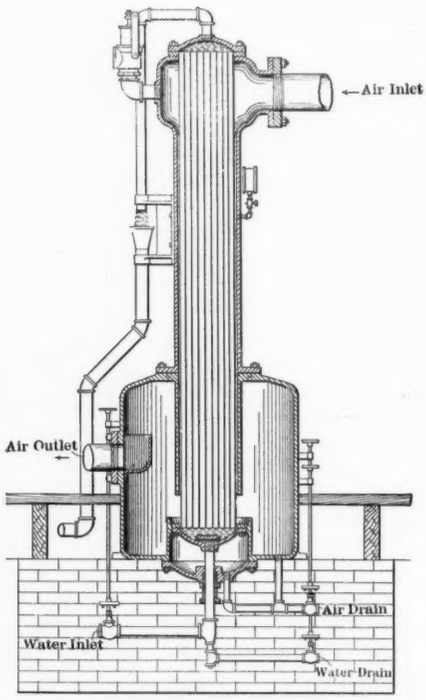
Vertical Intercooler for Air Compressors

The accompanying illustrations show the latest, and standard, vertical intercooler (used also as an antecooler and an aftercooler) of the Ingersoll-Rand Company. The air enters at the top, strikes across and among the thin brass cooling tubes and passes down the hexagon body, with only space sufficient for its free passage, thus securing intimate contact with all the tubes. The size and shape of the shell are continued by a petticoat down into the large settling chamber, and the air leaves the tubes near the bottom, circulates slowly around, drops its liberated water and passes out by the upward looking elbow at the side.

The water enters at the bottom and overflows at the top, keeping the tubes always full and avoiding all pressure, except the weight of the water, so that the tubes are very thin and highly efficient for their purpose. The upper head, in which the tubes are fastened, is held securely by the cap bolted over it; the lower tube head is free to slide vertically to avoid all strains of expansion or contraction, the water pipe being led in through a stuffing-box.

The tubes may be taken out without moving the body, and if the joint is broken at the top of the settling chamber not much hight is required for the lift. The capacity of the lower chamber gives it considerable value as an air receiver, although it is not intended that the usual air receiver shall be dispensed with.

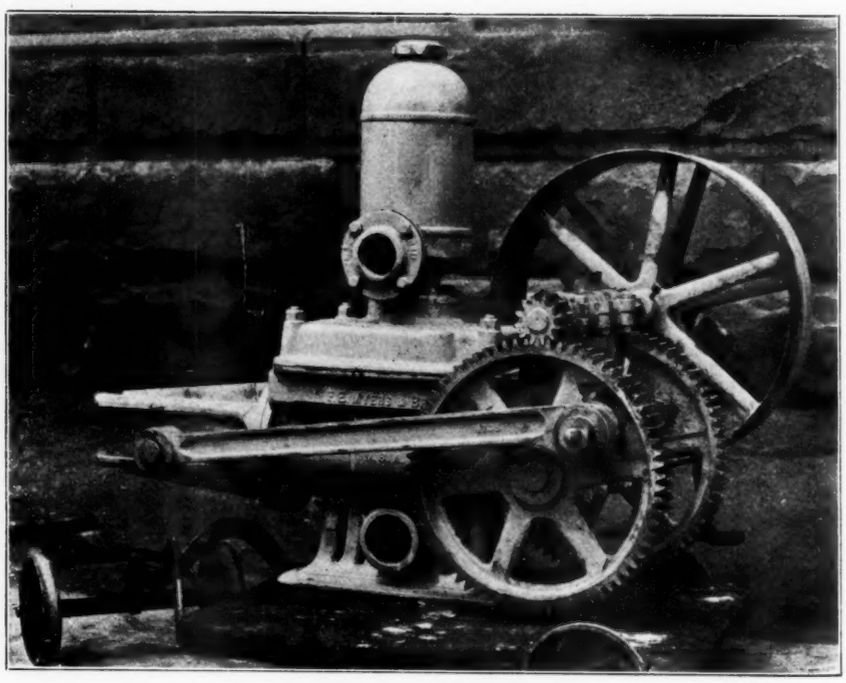




INGERSOLL-RAND VERTICAL INTERCOOLER

The Myers "Bulldozer" Power Pump

The pump illustrated in the accompanying engraving has been brought out by F. E. Meyers & Brother, of Ashland, Ohio, to meet the demand created by the electric motor and internal-combustion engine. The cylinder is formed by the main body of the pump, which is bored and lined with a heavy seamless-drawn brass tube 6 inches in diameter and of a length sufficient for a 10-inch stroke. The valve seats are of the grid pattern, fitted with hard-rubber valves 4½ inches in diameter, backed with coil springs. A connecting-rod is used on either side of the pump, insuring against displacement of the crosshead or deflection of the rod. The arrangement of the crosshead upon one side of the pump and the gears upon the other gives compactness without sacrificing accessibility. The gears are double, with 2½-inch face, extra heavy teeth, back-geared 6 to 1. The capacity of the pump illustrated at 40 strokes per minute is 5800 gallons per hour, and it will operate against a 150-foot head.



MYERS "BULLDOZER" POWER PUMP

Alcohol Distilleries on German Farms

As a result of recent investigations by Commissioner of Internal Revenue Yerkes and Representative E. J. Hill, of Connecticut, it was found that Germany is far ahead of America in the matter of applying denatured alcohol for the development of industrial purposes, there being already in that country 70,000 farm distilleries, many of them very small.

The stills have to be made in a certain way, says Representative Hill, which includes a tank that can be locked with a government lock and sealed with a government seal. The small farm distilleries do not operate all the year round. They operate in the winter, when the farmer has leisure to do something other than straight farm work. The farmer has to give the government 30 days' notice as to the time he wants to begin to operate his still. Some time during the 30 days an inspector comes along and looks the still over to see that it is clean, etc., and then he locks and seals the tank, after which the still is ready for the farmer.

Universal Peace and Commerce Exposition

Senator Flint has introduced a bill at Washington for the holding, in 1915, of an exposition (to be known as the Universal Peace and Commerce Exposition) at Los Angeles, California, to commemorate the opening of the Panama canal; and to include exhibits of the arts, industries, manufactures, and products of the soil, mine, and sea.

Book Reviews

"Stationary Engineering." By Joseph G. Branch, B. S. M. E. Perrin & Smith Printing Company, St. Louis. Cloth; 940 pages, 5x7½ inches; 300 illustrations. Price, $3.50.

This book is designed especially for the practical man, and the author has endeavored, by simple language, liberal illustrations and the avoidance of mathematics, to set forth the principles of steam engineering in a manner in which they may be understood by the man whose training has been practical rather than technical. In addition to purely steam-engineering subjects, chapters are devoted to electricity and magnetism, dynamos and motors, elevators, mechanical refrigeration, etc. Every chapter is followed by a list of questions enabling the reader to examine himself upon the matter which he has just read. The author is a member of the Board of Examining Engineers for the city of St. Louis, and has evidently derived from his practical contact with the men whom he has examined an idea of what they need in the way of instruction and the manner in which it can be best presented for their assimilation.

"Producer Gas and Gas Producers." By Samuel S. Wyer. Hill Publishing Company, New York; 1906. Cloth; 296 pages, 6x9 inches; 112 illustrations; numerous tables. Price, $4.

This is a most comprehensive and practical treatise on the subjects indicated by its title. The author's diction is bad in spots and his treatment of some sections of the work is too perfunctory, but his practical knowledge of his subjects is evident. His presentation, in the first two chapters, of the fundamental physical and chemical laws and definitions is somewhat marred by an attempt to be too concise, if the book is intended for a practical reference work as the character of the subsequent chapters seems to intimate. This is probably responsible for his definition of a perfect gas as one "which strictly follows Boyle's law." While the statement is etymologically accurate the phraseology is faulty in giving the impression that the performance of the gas is regulated by the law.

Chapter XI on the heat balance of the producer could have been expanded to double its length to advantage, and the chapters devoted to descriptions of commercial producers are also much too condensed and general.

On the other hand, the arrangement of the contents is good and the author's style of exposition is excellent, despite his tendency to extreme conciseness. From an engineering standpoint it is far superior to anything hitherto published in this country on the subject.

"Steam Turbines." By Carl C. Thomas. John Wiley & Sons, New York. Cloth; 287 pages; 5½x9 inches. Illustrated. Price, $3.50.

The principal incentive of the author has been the desire to analyze and correlate the results of experimental investigations, and to show how these results could be used in connection with the well known laws of hydraulics and thermodynamics as applied to steam turbines. The data used were obtained by Professor Gutemuth, of Darmstadt, Dr. Stodola, of Zürich, George Wilson, of Manchester, Walter Rosenhaim, of Cambridge, Professor Rateau, of Paris and Professor Carpenter, of Cornell, as well as by the author and others at Sibley College and at the Westinghouse, Allis-Chalmers and De Laval shops. The first three chapters are devoted to fundamentals. In chapters IV, V and VI the flow of steam through orifices and nozzles is discussed and experimental results compared with those obtained by calculation. The remainder of the book is devoted to the principles of turbine design and operation with the constant aim to show in what way the results to be expected may be predicted by the proper use of experimental data. A commendable feature is the manner in which the charts are inset so as to be accessible for reference at any opening.

"Electrical Engineering: an Elementary Text-book." By E. Rosenberg. Translated by W. W. Haldane Gee and Carl Kinzbrunner. Revised and brought down to date for the American market by Edward B. Raymond. Published by John Wiley & Sons, New York. Cloth; viii + 347 pages, 6x9 inches; 333 figures. Price $2 net.

Anybody can understand that which is written in this book—student, shopman or the layman seeking general information. The work is the outcome of a series of lectures which the author delivered before the employees and officials of a large electrical manufacturing company. It covers a wide scope, including fundamentals, dynamos and motors for direct and alternating current, accumulators, measuring instruments and electric lighting. Of course the fundamental phenomena of electricity have been so often and so simply treated that this work cannot be expected to differ greatly in that particular from what has gone before; nevertheless, it does simplify them somewhat, especially in the illustrations. One noteworthy feature is pointed out by the translator, viz., it is different from other elementary works on this subject in that it gives considerable space to alternating-current engineering, which is not treated of so mathematically as to be beyond the power of ordinary readers to grasp intelligently. In revising the work for American readers, certain subjects essential to American practice have been added.

"Boiler Waters: Scale, Corrosion, Foaming." By William Wallace Christie. Published by D. Van Nostrand Company, New York. Cloth; 235 pages, 6¼x9¼ inches; 77 illustrations. Price, $3 net.

This work is announced as "a pioneer on this subject in this country." The author proceeds on the assumption that "a steam boiler is a steam generator, not a kettle for chemical reaction," and his ideal deduction is that "the only compound to put into a boiler is pure water." As may be conceived, the proper treatment of such a vast subject involves the employment of more or less scientific data, especially of chemical properties, but the author has been most happily able to simplify the text so as to be readily interpreted by the engineer who has a superficial knowledge of such matters and wants to learn more. The first chapter contains a large fund of instructive information about water, together with analyses of river water, well water, sea water, etc., in different parts of the world. The section on chemical analysis, with directions for making up your own test outfit, and its application, should prove especially helpful. The chapter on boiler scale will surprise the reader by reason of its comprehensiveness, both text and illustrations. The rules for preventing scale and directions for its removal seem to be worth while, too. Corrosion is exhaustively treated of, also, and profusely illustrated. Feed-water pipes, blow-off pipes, and tubes, which are subject to harder usage than a good many engineers are apparently aware, are carefully considered and many examples of the results of deleterious neglect are cited and pictured. The causes of priming and foaming are tersely explained, and preventatives and remedies pointed out. One of the most important chapters is that on the troubles from the oil required to lubricate "oil-fed" engines, including also the presence of grease in boilers and the use of zinc to prevent boiler scale. The hardness of water, methods of ascertaining it, with tables, will likewise repay study. Feed-water heaters and water-softening systems are elaborately expounded, various types of each being described and illustrated. Tables of conversion, saving from heating feed-water, factors of evaporation, properties of saturated steam, expansion and weight of water, temperature of boiling, barometer and altitude, and chemical composition of substances, with symbols, conclude this interesting work, the whole being carefully indexed according to subjects.

The Engineering Quarterly of the University of Missouri made its initial bow in December. Like other college publications of this nature, it will be devoted to engineering and kindred sciences. The first number contains 80 pages, 6x9 inches, and is well put together.

Inquiries

Questions are not answered unless they are of general interest and are accompanied by the name and address of the inquirer.

Maximum Allowable Current for Induction Motors

I have a 60-horse-power induction motor that becomes very hot when the current per lead reaches 150 amperes; the line voltage is 200 volts. Is the machine overloaded at this current, or is the overheating due to some defect? C. J. F.

It is impossible to say positively whether or not your motor is overloaded without knowing the number of phases, the power factor and the efficiency of the machine. The full-load current per lead for an induction motor is computed as follows:

Two-phase motor:

$$\frac{373\ H.P.}{E\,e\,p} = I.$$

Three-phase motor:

$$\frac{430.7\ H.P.}{E\,e\,p} = I,$$

in which formulas E represents the voltage between terminals; e, the efficiency of the motor expressed as a decimal, and p the power factor also expressed decimally. Thus, if the full-load efficiency were 88 per cent., the value of e would be 0.88, and if the power factor is 90 per cent., the value of p is 0.9. If yours is a two-phase motor, taking a current of 150 amperes per lead at 200 volts, and if we assume that to be the full-load current, the product of the power factor and efficiency figures out

$$\frac{373 \times 60}{200 \times 150} = 0.746$$

which is a rather low value and would indicate that 150 amperes is an overload. If it is a three-phase machine, the product of p and e figures out

$$\frac{430.7 \times 60}{200 \times 150} = 0.8614,$$

which is a rather high value. The power factor and efficiency should be on the name-plate; if they are, apply the formula corresponding to your number of phases and you can readily see whether or not 150 amperes is an overload.

Gain Due to an Economizer

A boiler receives the feed-water at a temperature of 56 degrees Fahrenheit and changes it to steam at 132 pounds pressure gage. If the addition of an economizer to the plant raises the temperature of the feed of 223 degrees Fahrenheit, what is the percentage of gain? R. W. C.

A pound of water at 56 degrees contains 24.01 B.t.u. (a little more than a heat unit per degree, as the specific heat increases as the temperature goes up). A pound of water at 223 degrees contains 192 B.t.u. To convert a pound of water at 32 degrees into steam of 132 pounds gage, 147 absolute, requires 1190.7 B.t.u. The heat required to make the steam from the feed-water is for the two cases:

At 56°	At 223°
1190.70	1190.7
24.01	192.0
1166.69 B.t.u.	998.7 B.t.u.

The saving would therefore be:

1166.69 — 998.7 = 167.99 B.t.u., which is:

$$\frac{167.99 \times 100}{1166.69} = 14.39 \text{ per cent.}$$

Prony Brake Formula

What is the rule or formula for using a Prony brake? C. S

Multiply the length of the arm, in feet, from the center of the shaft to the point where the weight or spring is attached, by the weight or pull of the spring balance; multiply this product by the revolutions per minute, and multiply this last product by 1904 and divide the result by 1,000,000. The final result will be horse-power. In the shape of a formula:

$$\frac{1904 \times \text{arm length} \times \text{weight} \times \text{r.p.m.}}{1{,}000{,}000} = \text{H.P.}$$

Sizes of Pump Cylinders

Will you give me a good method of figuring the sizes of steam cylinders for simple, compound and triple-expansion straight-line duplex pumps when the capacity and lift are known? G. W. B.

Multiply the cross-sectional area of the pump plunger by the water pressure per unit area; divide the product by the lowest steam pressure in the cylinder with which the pump will be expected to run and the result will be the area of steam piston which would produce a balance between the steam and water pressures. Increase the steam area sufficiently to allow for the friction of the pump itself and of the water. This will vary with circumstances and is a matter of judgment. If too much allowance is made it can be compensated for by throttling.

Where steam is carried full stroke in the first cylinder, as in the ordinary duplex pump, the number of expansions is equal to the cylinder ratio in the case of a compound and the product of the succeeding cylinder ratios in the case of a triple. In the case of the compound it is obvious that the full volume of the high-pressure cylinder is discharged into the low-pressure cylinder so that the steam has been expanded to four times its original volume, for instance, if the low-pressure cylinder is four times as large as the high. If there is another cylinder the number of expansions in that cylinder will be its volume divided by the volume of the intermediate. If it be four times as large as the intermediate, for example, the steam delivered to it from the intermediate cylinder will be expanded four times and as it has already been expanded four times it will have $4 \times 4 = 16$ times its original volume.

The determination of ratios depends therefore upon how many times it is desired to expand the steam, and how it is desired to distribute the work among the cylinders. Each case must be worked out for itself.

Rate of Water Discharge under Diminishing Head

How can I figure the length of time theoretically required to empty a tank of water through an orifice of known size, the tank being a simple cylinder of the same diameter at all levels and the orifice being in the bottom of it? T. G.

If you represent by the letter f the amount by which the head would be diminished under the influence of a pressure equal to that of one-foot head; the head of water in the tank at the beginning of the flow by H, and the time in seconds required to empty the tank by t, the formula is

$$\frac{2\sqrt{H}}{f} = t$$

The value of f may be determined theoretically by the formula:

$$\frac{0.62 \times a \times \sqrt{2g}}{144 \times A} = f,$$

in which a = area of orifice in square inches; g = force of gravity, and A = cross-sectional area of the tank in square feet. The $\sqrt{2g}$ is equal to 8.02.

Changing the Voltage of a Standard Dynamo

Can I get more than 250 volts from a 250-volt direct-current dynamo, and if so, what changes would be necessary? J. B. S.

You can get a slightly higher voltage by increasing the speed of the armature, but if you attempt too great an increase you will get into trouble with sparking at the brushes. An increase of 10 to 15 volts is safe. No changes are necessary except to drive the machine faster; multiply the present speed by the desired voltage and divide by 250, and the result will be the higher speed at which the machine must run.

Effect of Draft Tube on a Water-wheel

Why is it that in putting in hydraulic turbines the turbine is sometimes located at a level almost as high as the head water? Does not this affect its power? C. H. W.

In order to make the turbine more accessible or to facilitate driving the load from its shaft. When a turbine is located above the tail-race, as it always should be, a draft tube is provided which forms an air-tight connection from the turbine to the tail-race. The column of water in this draft tube exerts by "suction" a pull downward equal to the pressure that it would exert if it were located above the turbine.

Annual Meeting of the Marine Engineers' Beneficial Association

The National Marine Engineers' Beneficial Association held its thirty-second annual convention at Washington, D. C., January 21-26, the headquarters being at the Ebbitt house. There were 129 delegates present, and much good work was accomplished. The reports of the various committees show the organization to be in a most prosperous condition. During the week the National Marine Engineers' Supplymen's Association, in session at the Ebbitt house, entertained the engineers and their guests with trolley rides, theater parties, etc., which were heartily enjoyed.

The leading social feature was a smoker held in the ballroom of the New Willard hotel, at which many Government officials and officers of the Army and Navy were in attendance, as were also the delegates of the Master Pilots' Association, also in convention at Washington. The entertainment was in the form of a minstrel show, first-class in every respect, the entire company coming from New York City especially for this occasion.

On Friday an opportunity was taken advantage of to call on President Roosevelt at the White House. Mr. Roosevelt expressed his pleasure in meeting the party, and congratulated the marine engineers on their high standing in the commercial world, after which the President shook hands with each individual.

On Wednesday the marine engineers elected the following national officers for the ensuing year: W. F. Yates, president; W. J. Brady, Jr., first vice-president; J. R. Blanchette, second vice-president; G. A. Grubb, secretary; A. L. Jones, treasurer; W. B. Blaicher, F. J. Houghton and William Sheffer, members of advisory committee. The election of officers of the Supplymen's Association resulted as follows: Charles W. Martin, Jr., chairman; John L. McGilvray, secretary-treasurer.

Manhattan and Bronx N. A. S. E. Entertainment and Ball

The fifth annual entertainment and ball of the ten combined associations of the National Association of Stationary Engineers of Manhattan and the Bronx was held at the Grand Central Palace, New York, on January 30. As usual, the event was a big social success and was attended by many prominent engineers and supplymen. The entire corps of National officers was present, namely: R. D. Tomlinson, past president; T. N. Kelsey, president; Joseph F. Carney, vice-president; F. W. Raven, secretary; S. B. Forse, treasurer; D. J. Hussey, doorkeeper. A first-class vaudeville entertainment was followed by dancing.

Author's Correction

Frank Foster, author of the paper on "Condensers for Steam Engines," an abstract of which appeared in the December, 1906, POWER, informs us that in consequence of the paper having been given out for publication prior to correction, errata are apparent in the data respecting the gain to reciprocating engines from high vacua, and the best vacua for these engines. Thus, on page 769, the quantities "0.74 pound," and "0.38 pound," given as the saving with reciprocating engines, should be halved, while the quantities 200, etc., on the top side of Fig. 3, page 770, and Fig. 5, page 771, should be doubled. Mr. Foster further states that in his book on "Steam Turbine Design," in the chapter on condensers, these corrections have been made.

The International Correspondence Schools, of Scranton, Penn., have embodied the souvenirs of the fifteenth anniversary of the founding of the schools in a handsome volume of 156 pages containing illustrations of the buildings, portraits of the members of the administration faculty, speakers both at the anniversary exercises and at the banquet, and the papers and addresses in full, together with a list of the guests. It is beautifully printed on heavy plated paper, and will be a welcome reminder of an inspiring occasion.

In commenting upon our recent editorial upon lapped-joint boiler seams, the Fidelity and Casualty Insurance Company says: "It is suggestive in this connection that a year or two ago a boiler maker from England came to us and tried to induce us to enter the boiler-insurance field in Great Britain. When we told him that the home companies could do the business to better advantage, he frankly stated that they would not insure boilers made on the lapped-seam principle, that he knew the American companies did, and that he wanted to make lapped-joint steam boilers."

The Supreme Court of Indiana has reversed a decision of a lower court by which a workman employed by a steel company secured damages of $2500 because he was scalded by steam turned on a boiler while he was inside cleaning it. The court held that the company was not shown to be to blame for the steam being turned on, and that it might have been due to the fault of a fellow workman, for whose acts the company was not liable.

Pending decision on the defendant's application for non-suit, the jury in a Toronto, Ont., court on January 16 awarded damages to the amount of $4500 to a former engineer of the Ontario Sewer Pipe Company, to compensate him for injuries received by being scalded by the bursting of a steam valve.

Personal

A. V. Pope has been appointed manager of the gas and electrical departments of the Birmingham Railway, Light and Power Company, Birmingham, Ala.

Charles A. Munroe, Chicago, Ill., has been elected general manager of the Economy Light and Power Company, of Joliet, Ill., to fill the vacancy caused by the death of Robert L. Allen.

John E. Zimmermann, formerly secretary of the American Pulley Company, of Philadelphia, Penn., has been admitted to partnership in the firm of Dodge & Day, engineers, also of Philadelphia.

On February 1, Henry Docker Jackson, consulting electrical engineer, removed his office from No. 4 State street, Boston, Mass., to room 626, Broad Exchange Building, 88 Broad street, Boston.

William B. McVicker, of the Dearborn Drug and Chemical Works, Chicago, is making an extended trip in Florida and Cuba, in which localities the company has lately built up large business connections.

W. H. Owens, superintendent of the Madison County Light Company, operated by the McKinley syndicate, with plants in Edwardsville and Granite City, Ill., has been transferred to Decatur, Ill. He was succeeded by E. D. Bell, of Granite City.

Obituary

William L. Simpson, a well known consulting mechanical engineer, died at his home in Philadelphia on January 27, at the age of 60 years. He was the founder of the Keystone Engine and Machine Works, which he conducted several years, and was at one time in the employ of the Southern Mail Steamship Company, being engineer of that company's steamship "Juniata," which foundered off the Florida Keys about 30 years ago. Mr. Simpson was prominent as a Mason, clubman and member of the American Society of Mechanical Engineers.

H. C. Weber, superintendent of the Weber Gas Engine Company, of Kansas City, died at Phœnix, Arizona, Monday, January 21 last, the immediate cause of death being a severe cold contracted while on a hunting and fishing trip in the Ozark mountains in southern Missouri last fall. Mr. Weber was born at Boonville, Mo., October 30, 1876. When 18 years old he began an apprenticeship in the machine shop of his father, Geo. J. Weber, and at 25 years of age he was appointed general superintendent of the gas-engine company. He supervised the installation of the first large gas-power plant in Mexico. Deceased was a 32d degree Mason and Mystic Shriner.

Business Items

The Austin Separator Company, Detroit, Mich., has sent us a very pretty desk calendar for 1907.

The Pittsburg Feed Water Heater Company, of Pittsburg, Penn., reports that recent sales of its heaters aggregate 6475 horsepower.

The new steel foundry of the Crane Company, Chicago, is now in full running order. In this department steel valves and fittings will be made a specialty.

The York Manufacturing Company, of York, Penn., manufacturer of ice-making and refrigerating machinery, reports recent sales amounting to 1154 tons of refrigeration.

The Curtis & Curtis Company, of Bridgeport, Conn., manufacturers of the "Forbes Patent Die Stock," expects to enlarge its plant in the spring. When completed, this will provide 6000 feet additional floor space.

The Gibson Iron Works Company, Jersey City, N. J., reports that it has just received an order for twelve sets of its patent common-sense dumping grate for the Northern Texas Traction Company, Fort Worth, Texas.

C. H. Mosher has been appointed eastern representative of the American Steam Gauge and Valve Manufacturing Company, of Boston, Mass., to succeed R. M. Turner, who was recently made manager of the company's publicity department.

Ilmer & Co., Cincinnati gas-engine engineers, are making a specialty of high-power gas-engine designs, thereby eliminating drafting-room work by the builders. They also make plans and estimates, and supervise gas-engine installations, etc.

Robert S. Wiley has taken over the control of the American Ship Windlass Company, Providence, R. I., which is improving its facilities in anticipation of a largely increased business, and is also preparing to push the development of the Taylor gravity under-feed stoker.

Charles K. Thomas, formerly general sales agent of the Lunkenheimer Company, has been appointed eastern sales agent of the D. T. Williams Valve Company. His headquarters are at 904 Broadway, Cincinnati, O. Mr. Thomas is also secretary of the National Exhibitors' Association.

M. Woolsey Campau will represent the C. C. Wormer Machinery Company, of Detroit, principally traveling in Michigan as salesman for steam-plant machinery and machine tools. Mr. Campau, who is a graduate of the University of Michigan, has been in the machinery business several years.

A commission appointed by the French government to look into petroleum (petrole lampant) engines for the French navy has recommended the Mietz & Weiss system, and has purchased the 22-horse-power marine engine of this type, which the commission had under test, and which was built by A. Meitz, 128 Mott street, New York.

The Direct Separator Company, of Syracuse, N. Y., recently contracted with the following to handle it specialties: Dashiell Engineering Company, 328 Candler Building, Atlanta, Ga., agent for the South; J. J. Dwyer, 317 American National Bank Building, St. Paul, Minn., agent for the West; A. C. Stansill, 404 Terry Building, Richmond, Va., agent for Virginia.

The Buckeye Boiler Skimmer Company, of Toledo, Ohio, has received from the Defiance (Ohio) Machine Works a good letter regarding its skimmers, of which the following is an extract: "We are delighted with the several skimmers which you connected with our boilers. We are using two equipments in connection with the two 100-horse-power boilers. They keep our boilers clean."

E. H. Kellogg & Co., 243 South street, New York, have received a letter from the chief engineer of the Denholme woolen mills at Denholme, England, in which he says that he considers the Kellogg "Anti Corrosive" cylinder oil the best on the market, having used it for over twenty years with the best possible results, never having found it to clog the pipes or coat the boiler plates.

At the officers' and branch managers' convention of the Crocker-Wheeler Company, held January 23 to 26 at Ampere, N. J., it was announced that the company had done more business during 1906 than any other year since its foundation eighteen years ago. Among the specialties that have contributed largely to the record made last year are alternating-current generators of large capacity and variable speed direct-current motors for driving machine tools.

The Reagan Grate Bar Company, of Philadelphia, has recently sold $11,000 worth of Reagan grates to the Great Northern Paper Company, of Millinocket, Me. These are to replace mechanical stokers and are ordered after an exhaustive investigation, by the purchasing engineer, of the Reagan grates in use at the plants of the Boston and Northern Street Railway Company for the burning of bituminous coal, which is Mr. Reagan's forte.

Adam Cook's Sons, the only makers of Albany grease have received a letter from H. Schroeder, chief engineer of the Consumers Company, Chicago, Ill., regarding his experience with this lubricant in which he says: "In regards to your Albany grease, there is nothing on the market that will beat it, for I had experience in using Albany grease for the last thirteen years; six years as oiler, five years as assistant engineer and two years as chief engineer.

The Sims Company, Erie, Penn., reports the year of 1906 as having been a very prosperous one in its line of business. The demand for larger units of Sims feed-water heaters is constantly growing. The company has just shipped a 2000-horse-power closed-type feed-water heater to the Western Penitentiary, Allegheny, Penn. The prospects for 1907, the manager says, are very flattering. The Sims people anticipate enlarging their plant in the spring to meet the growing demand for their steam appliances.

The B. F. Sturtevant Company, of Boston, Mass., has recently received orders for its new type of high-pressure rotary blowers from the General Fire Extinguisher Company, Atlanta, Ga.; Rivett Lathe Company, Brighton, Mass.; Wayne Agricultural Works, Goldsboro, N. C.; J. B. Clark Manufacturing Company, Rockford, Ill.; New York Continental Jewel Filtration Company, New York City; Winchester Repeating Arms Company, New Haven, Conn.; Salem Glass Company, Salem, N. J., and Eastern Chemical Company, Boston, Mass.

The H. W. Johns-Manville Company is popularly associated in the public mind with pipe coverings and kindred products. This has led them naturally into the manufacture of insulating materials and this into overhead line material, electric railway supplies, etc. One of the articles which they produce and of which they have said comparatively little is the National steel tube cleaner, consisting of four scrapers mounted upon long spring-like blades with guides to lead them into the tubes. It can be adjusted to fit various sizes of tubes, and if one or more of its blades become broken by rough usage or wear, they can be replaced at nominal expense.

In one of the branch factories of the Garlock Packing Company there is a left-handed man who works in the gasket department. He has to have special left-handed shears to do the cutting with and so the head office at Palmyra, N. Y., had made one dozen pairs to hold in stock and forward as needed. A new clerk in the head office did not know of these circumstances and when some came in and told him to send out an order to send one pair of left-handed shears to..... branch office, he replied: "Sure, I'm green. I suppose that stove-pipe wrench goes on that same order doesn't it?" It was about fifteen minutes' work for several people to convince him that he was not being made the victim of a joke.

The Canton-Hughes Pump Company, Canton, Ohio, successor to The Canton Pump Company, and The Snider-Hughes Company, of Cleveland, Ohio, has just completed the installation of machinery, received through the purchase of The Snider-Hughes Company, and also the erection of a two-story addition to their plant. This now gives the new company a machine room 300 feet in length. The stock room, pattern room and paint room occupy the two-story addition. The office has also been enlarged to take care of the increased business of the firm. The machine room is equipped with every modern device in the way of machinery especially adapted to the manufacturing of steam pumps quickly and economically.

New Equipment

The citizens of Elba, Neb., are considering the installation of an electric light plant.

Kentucky Electric Company, Louisville, Ky., has purchased site for a new plant.

The city of Farley, Ia., is said to be considering the installing of an electric light plant.

It is reported that the Marion (Ind.) Lighting and Heating Company will extend its plant.

The Iowa Pump Manufacturing Company, Fort Dodge, Iowa, is building an addition to its plant.

David Lay, of Red Lodge, Mont., is engineer for the proposed electric light plant at Joliet, Ill.

A new power house is being erected for the Bruer Brothers Lumber Company, Minneapolis, Minn.

The city of Ashland, Wis., will vote on the question of a municipal electric light and power plant.

The Hattiesburg (Miss.) Traction Company is contemplating the erection of a new power plant.

The Town Council, Stonewall, Miss., is considering the installation of an electric lighting plant.

The Stevens-Duryea Company, Chicopee, Mass., is erecting an addition to its automobile factory.

The city of Grottoes, Va., is securing preliminary estimates on the installation of an electric lighting plant.

The Milwaukee (Wis.) Refrigerator Transit Company will erect new shops to manufacture refrigerator cars.

The Argenta (Ark.) Light and Power Company has awarded contract for the erection of a new power house.

The General Electric Company, Schenectady, N. Y., will build an addition, 800x150 feet, to its turbine department.

A new packing plant, to cost $50,000, will be erected for the East Liberty Home Dressed Meat Company, Pittsburg, Penn.

The Town Council of Lake Odessa, Mich., is said to be considering the establishment of a municipal electric light plant.

The United Electric Light Company, Springfield, Mass., is considering plans for an enlargement of its steam plant.

Plans are being prepared for enlarging the Perry street station of the Lowell (Mass.) Electric Light Corporation.

The Great Northern Power Company, Duluth, Minn., is negotiating for a site in Superior, Wis., on which to build a plant.

William F. Broderick, A. A. Davis, of Midland, Md., and others, will organize a company to establish an electric light plant.

The Danbury (Conn.) & Bethel Gas and Electric Light Company is planning to make extensive enlargements and improvements.

The Hennessey (Okla.) Electric Light, Ice and Power Company has been organized and will build a plant. Fred Ehler, president.

The Latrobe (Penn.) Ice and Provision Company has awarded contract for a $50,000 ice and cold storage plant at West Latrobe.

A franchise for a gas plant at Boyne City, Mich., has been given to W. H. Selkirk, of Boyne City, and O. S. Heyden, of Petoskey.

J. T. Bebeau, of Mount Sicker, B. C., has under consideration the matter of installing an electric light plant at the Brenton Hotel.

The citizens of Davidson, Mich., have voted to issue $10,000 in bonds for the construction of a municipal electric lighting plant.

The capacity of the municipal light and power plant at Marquette, Mich., will be increased by the addition of another powerhouse.

The Brooklyn Edison Company has ordered a 6000-kilowatt Fullagar turbine from the Allis-Chalmers Company, for the Gold street station.

The Merchants Cold Storage and Ice Manufacturing Company, Richmond, Va., will erect a new boiler house at a cost of $10,000.

Plans are being made by the Detroit Edison Company for an addition to its plant at Delray, Mich., which will nearly double its capacity.

The Polar Wave Ice and Fuel Company, St. Louis, Mo., is preparing to erect a new ice plant which, with machinery, will cost $100,000.

The Brooklyn Borough Gas Company has purchased a site on Coney Island, N. Y., on which it is said a $1,000,000 gas plant will be erected.

The Union Bag and Paper Company, Sandy Hill, N. Y., is contemplating the construction of a hydro-electric plant and dam at a cost of $250,000.

The Salem (N. Y.) Light, Heat and Power Company has been incorporated; capital, $50,000. John F. O'Brien, of Plattsburg, is one of the directors.

The Van Auken-Clevaue Company, of Yonkers, N. Y., manufacturing steam specialties and marine engines, will erect a factory in Bridgeport, Conn.

The Orange County Traction Company, Newburgh, N. Y., will erect a new building to be used as a machine and repair shop, as well as car barn.

The purchasing department of the Edison Electric Illuminating Company, 360 Pearl street, Brooklyn, N. Y., would like to receive catalogs for its files.

Preparations are being made to rebuild the electric light plant at Caseto Rock, Wash., which was destroyed recently. H. D. Gates of Hillsboro, Ore., is owner.

Plans for improvements are contemplated by the Scranton (Penn.) Railway Company involving an expenditure of $585,000. These will include a new power house.

It is said that the Baptist Sanitarium Association, Dallas, Texas, will build a $100,000 power house in connection with sanitarium buildings now being erected.

New Catalogs

General Electric Company, Schenectady, N. Y. Catalog. Railway supplies. Illustrated, 233 pages, 8x11 inches.

Advance Packing and Supply Company, Chicago, Ill. Catalog. Packings. Illustrated, 39 pages, 3½x6 inches.

The Bruce-Meriam-Abbott Company, Cleveland, Ohio. Catalog. Gas engines. Illustrated, 28 pages, 6x9 inches.

B. F. Sturtevant Company, Boston, Mass. Catalog No. 140. High-pressure blowers. Illustrated, 48 pages, 6½x9 inches.

Nason Manufacturing Company, 71 Fulton street, New York. Catalog "T." Steam traps. Illustrated, 8x10½ inches.

Revere Rubber Company, 70 Kingston St., Boston, Mass. 1906 catalog. Packings. Illustrated, 27 pages, 6½x9½ inches.

De La Vergne Machine Company, foot East 138th street, New York. Folder. Klein water-cooling tower. Illustrated, 4 pages.

Joseph Dixon Crucible Company, Jersey City, N. J. Pamphlet. Air-compressor lubrication. Illustrated, 22 pages, 6x9 inches.

G. C. St. John, 143 Liberty street, New York. Catalog. St. John indicating and recording steam meter. Illustrated, 6½x9 inches.

Jacobson Machine Manufacturing Company, Warren, Pa. Bulletin "A." Jacobson gas and gasolene engines. Illustrated, 7x10 inches.

The Loew Manufacturing Company, Cleveland, Ohio. Catalog. Loew boiler-feed-water heaters. Illustrated, 31 pages, 5½x6½ inches.

Cleveland Rubber Works, Mechanical Rubber Company, Cleveland, Ohio. Catalog. Marco packings. Illustrated, 60 pages, 5x8 inches.

The Wm. B. Pierce Company, 327 Washington street, Buffalo, N. Y. Catalog. The "Dean" boiler-tube cleaner. Illustrated, 31 pages, 5x8 inches.

The Ohio Injector Company, Wadsworth, O. Catalog. Injectors and ejectors, sight-feed lubricators, grease cups, oil cups. Illustrated, 68 pages, 6x9 inches.

Help Wanted

Advertisements under this head are inserted for 50 cents per line. About six words make a line.

SALESMAN—An experienced man by New York house to sell high class steam specialties. Give particulars. Box 203, POWER.

SALESMAN WANTED for the sale of condensers. Schutte & Koerting Co., 12th & Thompson Streets, Philadelphia, Pa.

WANTED—Good gas and gasolene engine salesmen. State ability, experience, etc., in first letter. Atlas Engine Works, Indianapolis, Ind.

WANTED—Experienced water tube boiler erectors; permanent positions with the principal manufacturer of the United States. Address Box 193, POWER.

WANTED—Analytical chemist, thoroughly versed in preparing boiler compounds to handle scale forming properties found in water. Box 207, POWER.

WANTED—High-class designing engineer for rotary valve engines. Salary $5000 to $10,000 per year to man who has had definite ly successful experience. Box 191, POWER.

CASUALTY COMPANY invites applications from competent mechanics experienced in boiler and elevator work for the position of field inspector; and from engineers having in addition an intimate knowledge of general factory equipment and operation, for the position of chief inspector. Casualty, 209, POWER.

ENGINEER DESIGNER, familiar with American and European practice in large gas engine and gas producer work; capable of independent action as a designer of large, heavy-duty combustion engines, and to take entire charge of such work. State qualifications, salary and when available. Box 198, POWER.

WANTED—A concern manufacturing mechanical rubber goods require, for their steam packing department, an energetic salesman, who has had some experience in selling steam and oil packing, gaskets, etc., to the trade. Address giving full particulars, as to previous experience, with references; also, state compensation required. Packing, 202, POWER.

Situations Wanted

Advertisements under this head are inserted for 25 cents per line. About six words make a line.

ENGINEER, producer gas engine power and electric plants. Large experience in design, manufacture and installation. Box 208, POWER.

YOUNG MAN at present employed as chief engineer, desires change; would like position with a reliable engineering firm. Box 206, POWER.

WANTED—By young engineer, position as assistant to chief engineer or purchasing agent in large power plant; M. E. degree in '06; now with contracting engineers. Box 200, POWER.

CHIEF ENGINEER of large steam electric plant wishes to change; practical and technical education; expert with indicator and valve gears; excellent references; all letters answered. Box 178, POWER.

ENGINEER—Nine years' experience in construction, erection and operation of large steam and electric power plants, also 2 years' construction, erection and operation large gas power plants, desires position consulting or superintending. Address Box 201, POWER.

ENGINEER DESIRES to make a change; 25 years in large plants; 16 years in one place; institution or public building preferred; highest references; absolutely reliable; first-class man; understands electricity, plumbing and steam fitting; age 45. Address Box 199, POWER.

WANTED—Connection, preferably with prominent steam or electrical engineeering or machinery manufacturers to introduce their goods or place agents throughout South Africa, Australia, China, Japan, India, Java, Europe or Asia; have most valuable connection in these countries; can furnish satisfactory reference; have established agencies for one of largest export firms in America and Europe. Address "R. E. M.," 113 Bank St., New York.

Wanted—To correspond with firm (reliable), any city in United States not less 10,000 population; mechanical, electrical engineer; age 31 years; twelve practical with steam, electric, D. C. & A. C., 3 phase, etc.; high-speed automatic and Corliss condensed and simple; chief engineer of plants ranging to 2000 h.p., all modern auxiliaries. Wish to have charge electric or water plant, or erect on road for engineering firm; references. Box 204, POWER.

Miscellaneous

Advertisements under this head are inserted for 50 cents per line. About six words make a line.

GREENFIELD VERTICAL ENGINES lead the market. Send for circular. Address Greenfield Engine Works, Harrison, N. J.

EXPORT AGENCIES WANTED for Great Britain in steel, iron, tools and machinery. W. J. Kershaw, 34 Waterloo St., Birmingham, England.

IF YOU DESIRE to learn the latest improvement in steam boilers correspond with the Detroit Water Tube Boiler Co. See their advertisement on page 165.

ENGINES AND BOILERS, ⅛ to 2-h.p. engine castings in sets. Model and general machine work. Sipp Electric and Machine Co., Paterson, N. J. Catalog 4c.

PATENTS—H. W. T. Jenner, patent attorney and mechanical expert, 608 F St., Washington, D. C. I make free examination and report if patent can be had, and exact cost.

PATENTS SECURED PROMPTLY—Highest references from prominent manufacturers. Write for Inventors' Hand Book. Shepherd & Parker, 512 Dietz Building, Washington, D. C.

C. L. PARKER, solicitor of patents, 604 F St., Washington, D. C. Patents secured promptly and with special regard to the legal protection of the invention. Handbook for inventors sent upon request.

ENGINEERS AND FIREMEN—Send 10 cents in stamps for a 40-page pamphlet containing a list of questions asked by an examining board of engineers. Stromberg Publishing Co., 2703 Cass avenue, St. Louis, Mo.

WANTED—Engineers, electricians, firemen, machinists, etc. New pamphlet containing questions asked by the different examining boards throughout the country sent free. Geo. A. Zeller Book Co., 36 So. 4 St., St. Louis, Mo. See page 95.

WE HAVE on hand a few bound volumes of POWER for the years 1902, 1903, 1904. We need the space they occupy, you need the books if you are interested in power plant engineering. In order to move them quickly we will send them to the men who send us $2 a volume; state the year you want. The purchasers pay express charges. When these volumes we have on hand are gone there will be no more. Money received too late will be returned the same day. POWER, 505 Pearl street, New York City.

For Sale

Advertisements under this head are inserted for 50 cents per line. About six words make a line.

FOR SALE—A slightly used engineering library, bound ¾ leather, 10 vols.; cost $50. Will sell for $15. Address B. R. T., Box 425, Chicago.

FOR SALE—One Reeves speed controller, class G, No. 1, 6¼ to 9 horse-power, excellent condition. Price $50. Fels & Company, Philadelphia, Pa.

FOR SALE—Cyclopedia of Engineering, 4 volumes, bound in half morocco, slightly used; cost $19. Will sell for $5. Address T. M., Box 425, Chicago.

FOR SALE—Cyclopedia of electricity; five volumes, three-quarters red morocco, leather binding; books are in fine condition; used but short time; cost $19; will sell for $9. Address "R. S. M.," POWER.

FOR SALE—A slightly used engineering library. Steam, electrical, mechanical. Twelve volumes bound in three quarters red morocco leather. Cost new, $60. Will sell for $18. Address "R. L. N.," POWER.

FOR SALE—Cyclopedia of engineering; steam, electrical, mechanical; bound in three quarters morocco; four volumes; cost new $19; are in fine condition; will sell for $9. Address "M. B. T., POWER.

FOR SALE—Factory site of 57 acres in Wilmington, Del., within ten minutes walk from Pennsylvania depot, three railroad connections, 1500 feet of water front on navigable river, no city taxes for ten years. Address E. Krell, Narberth, Pa.

FOR SALE—The controlling interest in an engineering works, with fully equipped machine shop, building steam engines and ice machines. Plenty of orders ahead. Location, Pennsylvania. A good opportunity for a mechanical engineer. Box 205, POWER.

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Northern Engineering Wks.	152
Nugent & Co., Wm. W.	142
Ohio Blower Co.	151
Ohio Injector Co.	104
Oil & Waste Saving Machine Co.	139
Olds Gas Power Co.	170
Oneida Steel Pulley Co.	146
Oswego Boiler & Engine Co.	167
Otto Gas Engine Works	172
Owens, Isaac B.	101
Parmelee Wrench Co.	144
Patterson, D. W.	160
Patterson & Co., F. L.	157
Pearson & Co., J. H.	114
Peerless Rubber Mfg. Co.	105
Penberthy Injector Co.	103
Pennsylvania Rubber Co.	115
Philadelphia Gear Works	146
Philips Pressed Steel Pulley Works	149
Phosphor Bronze Smelting Co.	116
Pierce Co., Wm. B.	150
Pilley Packing & Flue Brush Mfg. Co.	149
Pittsburgh Gage & Supply Co.	139
Pittsburgh Valve & Fittings Co.	124
Pittsburgh Valve, Foundry & Construction Co.	120
Platt Iron Works	164
Platt & Co., John	142
Plymouth Cordage Co.	88
Potter Separator Co.	136
Powell Co., Wm.	101
Power & Mining Machinery Co.	172
Power Plant Specialty Co.	134
Power Specialty Co.	167
Providence Engineering Wks.	173
Quaker City Rubber Co.	113
Reagan Grate Bar Co.	132
Reliable Water Gauge Co.	132
Reliance Gauge Column Co.	143
Revere Rubber Co.	90 and 91
Rhoads & Sons, J. E.	147
Richardson Scale Co.	133
Ridgway Dynamo & Engine Co.	175
Robb-Mumford Boiler Co., Inc.	166
Robertson & Sons, Jas. L.,	108 and 109
Rollins Engine Co.	176
Ruboil Belting Co.	149
Russell Engine Co.	177
Ryan, E. J.	143
St. John, G. C.	116
St. Louis Iron & Mach. Wks.	177
Salamander Grate Bar Co.	130
Sargent Steam Meter Co.	116
Sayen, Osgood	95
Scaife & Sons Co., Wm. B.	144
Schaefer & Budenberg Mfg. Co.	140
Schieren & Co., Chas. A.	79
Scully Steel & Iron Co.	149
Sherwood Mfg. Co.	139
Shultz Belting Co.	148
Sight Feed Oil Pump Co.	139
Sims Co.	160
Skinner Engine Co.	173
Smith Gas Power Co.	170
Smooth-On Mfg. Co.	106
Spacke Machine Co., F. W.	160
Star Brass Mfg. Co.	125
Starrett Co., L. S.	126
Steel Mill Packing Co.	117
Stephenson Mfg. Co.	148
Stewart Heater Co.	157
Stocker, Geo. J.	161
Stow Mfg. Co.	140
Strong, Carlisle & Hammond Co.	141
Struthers-Wells Co.	172
Sturtevant Co., B. F.	154
Swain Lubricator Co.	117
Thompson & Co., Richard	149
Thompson & Sons Mfg. Co., J.	172
Trill Indicator Co.	140
Trimont Mfg. Co.	144
Tripp Metallic Packing Co.	116
Triumph Electric Co.	140

The Proper Management of Corliss Engines

Treating Especially of the Adjustment of the Valve Gear and How to Avoid Mistakes Often Made by Engineers

BY W. H. WAKEMAN

When an engineer takes charge of a Corliss engine for the first time, and must start it and keep it in operation for several days before he has an opportunity to take it apart and study its design and construction, I suggest that he first place the valve gear in the center of its travel, as illustrated in Fig. 1, although it will be necessary to unhook the valve-rod from the wrist-plate in order to do this, unless the engine happens to be standing in just the right position. The drawing illustrates the old-fashioned crab-claw valve gear formerly used on the Harris-Corliss and other engines, and having had much experience with this style, I am not ready to condemn it without qualification, as it wears well and does good work when properly cared for and not run at a high speed. Moreover, it illustrates the point that I wish to present, for when the wrist-plate is in the central position, as shown, the parts by which the steam valves are operated should stand in the same comparative position. If they do not, the reason for the difference ought to be found out as soon as possible, and if it is due to defective adjustment the fault can be corrected at the first opportunity. The same reasoning applies to the exhaust valves. Before an attempt is made to start the engine, however, the wrist-plate should be moved to its extreme travel in both directions, because if any of the parts are so adjusted that this is not possible, something might be broken by hooking on the valve-rod and turning on enough steam to give the fly-wheel one or more revolutions.

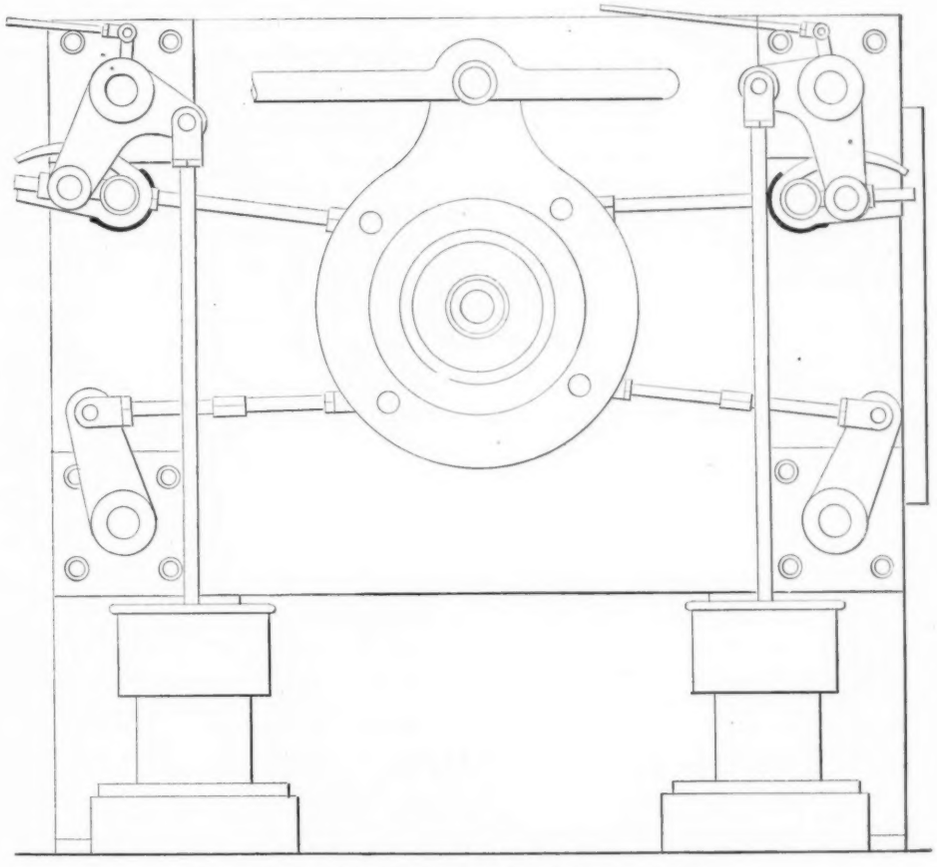


FIG. 1. VALVE GEAR SET IN THE CENTER OF ITS TRAVEL

Fig. 2 illustrates the proper way to tell when the wrist-plate is moved to its extreme travel. It is a plan of this part of the valve gear, and on the hub there is a line *2* which corresponds to the line *3* on the stud when the wrist-plate stands central. When *2* is moved over to *4* it is the extreme position in one direction, and when brought even with *5* it is the same for the opposite direction. These marks may not be exactly the same on all Corliss engines, but there is or should be something to give the same results.

Fig. 3 shows a different kind of a wrist-plate, but the principle involved is the same in all of them. In this case the dash-pot rod is too long, hence the wrist-plate will not rock to the extreme position in that direction indicated by the marks above described. Under these conditions if all other adjustments are properly made and the engine is started, some-

thing must bend or break, and it will probably be the dash-pot rod, especially if it is a long one, as in the illustration. This may have been lengthened purposely, in an effort to cause a new engineer trouble when he first starts the plant, by somebody who wishes to see him fail, or it may be caused by ignorance when making adjustments to compensate for wear as follows: When the valve gear of a Corliss engine is properly adjusted, the crab-claw, as illustrated in Fig. 3, slides past the block far enough to insure a good hold, but in course of time the wear on the several bearings causes the motion of the hook to be reversed before it engages the block, therefore the valve is not opened and the engine runs slower because it was not designed to carry its load while single-acting.

The engineer naturally wishes to apply a remedy as soon as possible; therefore, if he has not fully grasped the situation he may proceed to lengthen the connection without testing it, the rod will probably be bent, as already mentioned. While this would not disable the engine, it is disgusting to see the effects of such mistakes. If the dash-pot is located higher, making the rod short and stiff, the above-mentioned mistake may cause the bell-crank to be broken, as this part is usually made of cast-iron, and this will disable the engine (unless a spare one is at hand, which is usually not the case) as effectively as if the fly-wheel was broken. The proper method is to shut down the engine, be sure that the dash-pot is in its lowest possible position, then lengthen the rod until the crab-claw will hook on when the wrist-plate is rocked to its extreme travel in that direction (as above described), then hook on the valve-rod and start up again.

A Word of Caution

A word of caution will not be out of place here, for if a Corliss valve does not open when it should, it may be due to the fact that the edge of the block is worn until it will no longer retain the hook, but allows it to slip off immediately after it has hooked on; therefore, the valve does not move although the crab-claw hook, or whatever device is adopted for this purpose, travels far enough to accomplish its object. It may be possible to turn the block so that another corner will be available, and if it has not been used before it will do good work for a long time, provided the hook is properly ground. If all of the corners have been used until they are worn too much to hold, they must be refitted by grinding on an emery wheel or a grindstone, the latter being preferable on this account.

Do not stiffen the spring under a crab-claw in an effort to make it hold when it needs repairs, as that will make it close with a snap that is disagreeable and uncomplimentary. Furthermore, it requires more force to unhook it when enough steam has been admitted for that stroke, causing unnecessary wear.

Fig. 4 illustrates a mistake that is frequently made by engineers who are not familiar with this valve gear. Suppose that in a busy mill or factory the engineer suddenly receives orders to shut down, as a belt has run off from a pulley and is entangled in the machinery. His anxiety to bring the engine to rest as soon as possible and avoid stopping on the center causes him to unhook the valve-rod and reverse the wrist-plate while the crank is still moving fast enough to turn several revolutions more, and as there is a considerable momentum to overcome, he throws the wrist-plate forward with sufficient energy to carry it further than it ought to go.

The half-moon type of valve gear is illustrated here and it is assumed that when the wrist-plate was reversed it was moved from right to left, and after it had passed the limit described in connection with Fig. 2, the right-hand steam valve was disengaged, which caused the wrist-plate to go farther still in the wrong direction; therefore the bell-crank was reversed, and when the wrist-plate was pulled back again the steam-valve gear appeared as shown in Fig. 4. By this time the engine had stopped on the center,

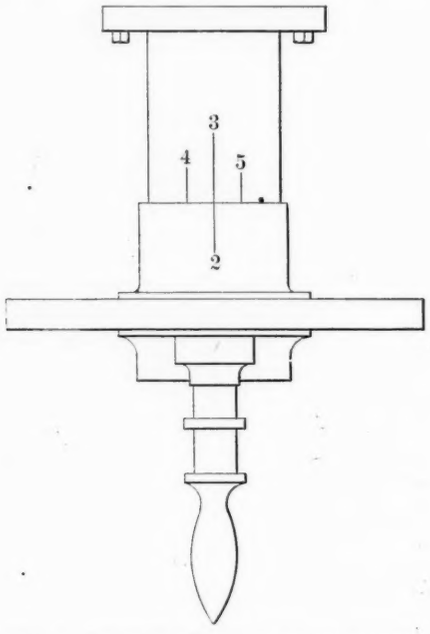


FIG. 2. WRIST-PLATE IN CENTRAL POSITION

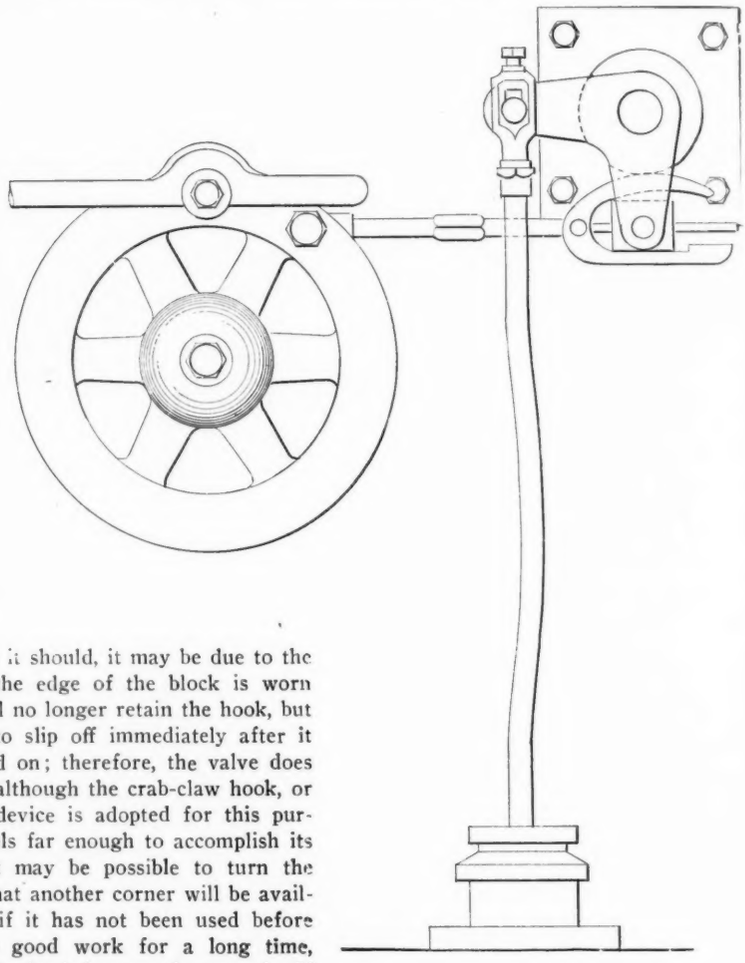


FIG. 3. DASH-POT ROD TOO LONG

between the wrist-plate and the valve crank until the hook engages the block, and the valve is opened properly as it appears to him; but what effect has this adjustment had on the setting of the steam valve? If it was set right before, it is now late in opening; consequently, steam does not get into the cylinder soon enough to give full pressure at the commencement of the stroke; therefore, the point of cut-off must be lengthened enough to give a diagram of equal area to the one secured before the change was made. This means a direct loss of steam due to an improperly set valve.

There is another way for correcting this defect, which consists of lengthening the dash-pot rod, and this is proper, because it does not change the set of the valve. This should be done cautiously, for if it is made too long and the engine is started

while several employees were watching proceedings and making facetious remarks, which added to the engineer's confusion. It is possible to hook on the valve-rod and make an attempt to start the engine, but it is not safe; for, although it would not run continuously in this condition, in some cases it will move far enough to bend or break something.

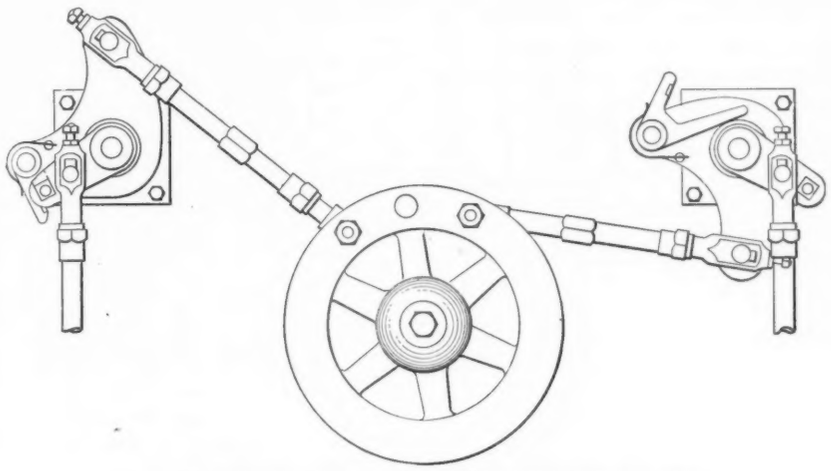


FIG. 4. SHOWING STEAM-VALVE MECHANISM TURNED OVER

If the right-hand bell-crank is moved back to the position in which the left-hand one appears, the engine will be ready for business, provided the wrist-plate was not thrown back far enough to turn one of the exhaust valves over, as indicated by the right-hand jim-crank illustrated in Fig. 5. In one case that was brought to my attention an electric-lighting engine was started in this condition and run for some time before it was discovered by the engineer. This is possible because the exhaust valve would remain open, and while this would allow steam to blow through to the exhaust chest and be wasted, it would not offer resistance to the advance of the piston, as it would in case the key which fastens the jim-crank to the valve stem would become loosened, work out and drop down on the floor, leaving the exhaust valve shut. In that case compression would rise high enough to be a serious matter, for it would probably stop the engine. In more than one case the engineer has found his Corliss engine slowing down from this cause; but by quickly inserting the key as the jim-crank came into proper position for it, normal speed has been restored and nobody else knew what caused the variation.

In the preceding illustration connection is made on the outside of the jim-crank, thus making it possible to turn the exhaust valve over, but Fig. 6 illustrates part of a valve gear in which connection is made on the inside; therefore, if the wrist-plate is thrown over too far, the small rod which drives the crank will strike the valve-stem bearing, possibly with force enough to bend the rod, but it will prevent the exhaust valve from turning over.

KEY TO TAKE UP LOST MOTION

Sometimes a key is used to take up lost motion at this point (see Fig. 6), and as the parts continue to wear, the key is driven down until the lower end of it strikes the valve-stem bearing, throwing it up again, and although it moves but a very short distance it is enough to make the box loose. Such a fault is not always readily discovered unless the engineer's attention is called to it by the experience of others along the same line, and although a remedy can be provided by cutting a piece off from the key, it is much better to put a liner behind the box and thus raise the key until it no longer rubs on the stationary bearing.

While it is generally necessary to unhook the valve-rod and reverse the wrist-plate when shutting down a Corliss engine, it should not be done when starting. It is perfectly proper to admit steam to each end of the cylinder, alternately, in order to heat it before starting, but the practice of rocking the wrist-plate back and forth in advance of the time it would be moved by the valve-rod, during several revolutions of the fly-wheel, is all wrong. Let us prove this by following the relative motion of the parts. Under certain conditions an exhaust valve will close when the piston is within six inches of the cylinder head when the valve-rod is hooked on the wrist-plate, and when it is not attached the engineer may close the exhaust valve when the piston is from twelve to eighteen inches from the cylinder head, but what advantage is gained by it? Water resulting from the condensation of steam is prevented from going out through the exhaust port; consequently, it takes longer for it to work out than it otherwise would, and it may not all go until the valve-rod is hooked on and the speed increased. If it was practicable to move the wrist-plate later than the valve-rod would carry it, there would be some advantage in this respect, but there cannot be as it is usually done; therefore, it is better to warm the cylinder, then hook on the valve-rod while the fly-wheel is still at rest, give the cylinder enough steam to turn the crank slowly for a few minutes, then gradually bring it up to full speed.

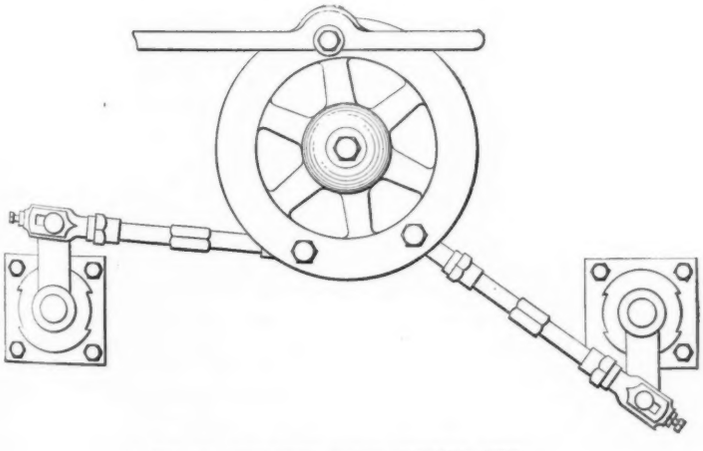


FIG. 5. EXHAUST VALVE TURNED OVER

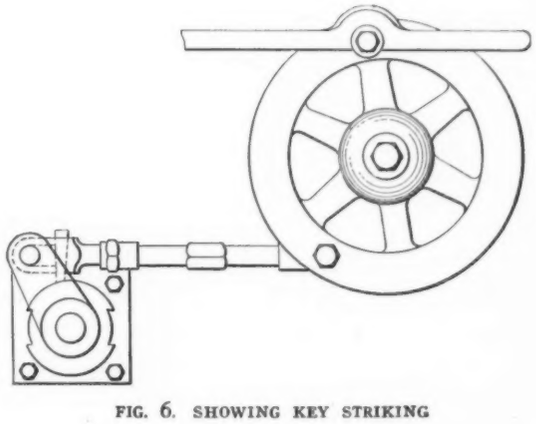


FIG. 6. SHOWING KEY STRIKING

If this is done carefully one valve will be tripped several times before the other is released at all. This will be right or wrong according to what is wanted of the engine. If it is to carry a heavy load, the tripping device should be set so that both valves will be released during one revolution, but if only a light load is to be put on, the valves ought to be released at the

same point of cut-off when the average load is on. Do not attempt to make the points of cut-off equal for both heavy and light loads, as the effort will probably result in disappointment.

The throttle valve ought to be opened wide enough to give full capacity of the steam pipe, in order to use steam to the best advantage, and give full power whenever it is wanted. If it is not opened enough the steam will be wiredrawn (which usually means a loss), and if more load is thrown on the speed will be reduced. There are many throttle valves in use that have not been opened wide enough to meet this requirement in a long time; consequently, an attempt to do it now would prove to be a hard job.

This does not necessarily mean that the wheel ought to be turned backward as far as it will go every time, as that might be too wide for convenience. When starting an engine it is no hardship to give the wheel a few more turns, but if an accident happens in the mill or shop, making it necessary to stop the machinery as soon as possible, the time that it takes to give those turns may make much difference in the results. Of course, this refers to cases where no special device is installed for the purpose of shutting the throttle valve. For illustration, take an ordinary 6-inch angle-valve as usually made for Corliss engines. When the valve has been run back one and one-half inches from its seat it is useless to give it more opening, while less will not give best results with a heavy load.

Valve gears of Corliss engines frequently wear rapidly for lack of proper lubrication, as many of them are oiled just before they are started, and are given no more attention for five hours. This is a great mistake, and after an engineer's attention has once been called to it, he should never allow it to take place again, as it is neglect of good machinery. In some engine-rooms these gears are thoroughly oiled every hour, and it would be done in others if the bearings were designed and made so that when one of them lacked lubrication it would melt and run down on the floor, but they are not; hence, when they are well oiled they run easily, and when dry they wear rapidly, but do not fail under the abuse. Cylinder-oil should be used on every bearing that is heated by steam, as ordinary thin oil is not suitable for such service. Many oil-cans are ruined in this work by attempting to follow small bearings that have peculiar motions, and thus get oil into small holes, but it would often be better to hold the nozzle still until the oil hole is under it at the extreme end of its travel.

A dozen 8000-kilowatt steam-turbine units have been under construction during the past year and four are in operation. These great engines will be used by the Edison lighting companies of Chicago, New York, Detroit and Seattle.

Reading Two-stage Air-compressor Indicator Cards

BY FRANK RICHARDS

The editor of POWER has handed me a letter in which a reader asks: "How do you figure from indicator cards taken from the cylinders of a two-stage air compressor the amount of air in cubic feet per minute, which will be delivered?" The writer also asks for "a diagram showing what constitutes a good card taken from the air cylinders of a two-stage air compressor." The inquiries suggest that the writer has some indicator cards from which he wants to get as much information as possible. If he would kindly send specimens of them to the editor with all the necessary data, I would certainly be glad to see them, as would probably others, and we might learn something from them.

As to the first question: The card from the first or low-pressure cylinder will, of course, show the volume of free air which is taken into the first cylinder, is compressed there and delivered or passed on to the next cylinder, and we may assume that this is a complete and sufficient record of the performance of the compressor so far as the volume of air is concerned; because all the air which passes the discharge valves, and is held so that none of it returns, must either leak out somewhere or must go through the high-pressure cylinder, and be compressed and delivered at the pressure required for service. It is assumed here that the compressor is in good working condition, and that the usual precautions have been taken to prevent leakage.

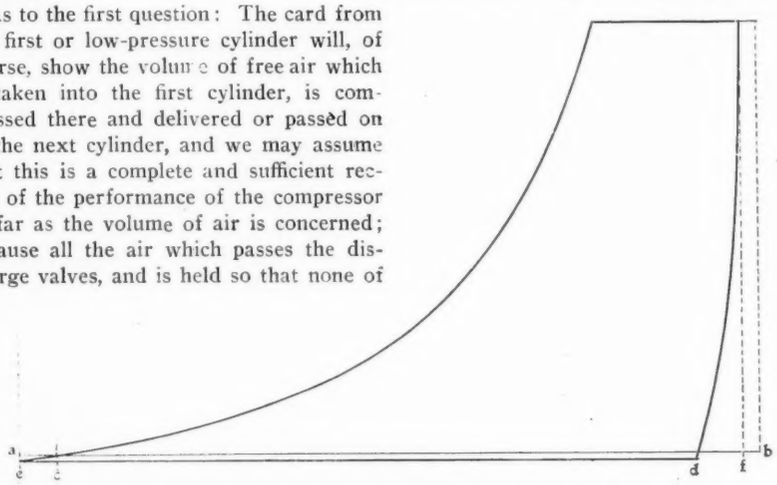


FIG. 1. TYPICAL AIR-COMPRESSOR CARD, SHOWING FREE AIR CAPACITY

Fig. 1 is a card from a single-stage compressor, but will serve our purpose as well as any. The card is idealized, or is smoother and sleeker than is obtained in practice, and the atmosphere line *a—b* is purposely drawn high enough to be entirely distinct from the air-intake line *e—d*. By "free air" in compressed-air practice is meant air at atmospheric pressure, and to be quite correct it should be the atmospheric pressure at the actual locality of the compressor, as the pressure varies with the altitude.

When the compression stroke begins, at *e*, if the air pressure in the cylinder is at all below that of the atmosphere we cannot, of course, take account of the volume of air in the cylinder until its pressure is brought up to atmosphere, or until the compression line coincides with the atmosphere line as at *c*. The compression then continues until the delivery pressure is reached, when for the remainder of the stroke the work of the piston is the expelling of the air from the cylinder. At the termination of the stroke, there being always more or less clearance space between the piston and the cylinder head, the air filling this space is not expelled or delivered, but upon the return stroke of the piston it re-expands and its pressure falls until atmosphere is reached at *d*, and, this being the intake stroke, the free air then, and not until then, begins to rush into the cylinder. As the total length of the line *e—f* represents the total cylinder content per stroke, the length *c—d* represents the free air actually delivered; and the ratio of *c—d* to *e—f* represents the percentage of free air actually compressed and delivered. The total length *e—f* of the card being 3.875 inches, and the free-air length *c—d* being 3.45 inches, the percentage of the full cylinder capacity actually compressed and delivered is $3.45 \div 3.875 = 0.89$. This is not at all intended to represent actual practice, as the best compressors show a better volumetric efficiency than this, but not all of them do as well as is generally claimed for them.

Instead of "a diagram showing what constitutes a good card from a two-stage air compressor" I offer here a pair of cards from an actual compressor completed and tested within a month or two, the cards giving record of the performance of what we may call a good compressor until someone puts us on track of a better.

The diameters of the air cylinders were, respectively, 32¼ and 20¼ inches, by 24 inches stroke; the diameters of the piston-rods of both cylinders were 3⅜ inches, and, both cylinders having the piston inlet, the outside diameters of the piston-inlet pipes were 10⅜ inches and 6 7/16 inches, respectively. The net piston areas for the low-pressure cylinder are:

Rear end (piston inlet), 732.32 square inches.

Forward end (piston-rod), 807.92 square inches.

The full cylinder capacity then per double stroke is:

$$(732.32 + 807.92) \times 2 \text{ (feet stroke)} \div 144 = 21.39 \text{ cubic feet,}$$

or at 80 revolutions per minute, the actual running speed when the cards here shown were taken, 1711 cubic feet per minute.

In Fig. 2 we have, full size, the cards from both ends of the low-pressure cylinder. They are reproduced as faithfully as possible, and in addition I have drawn a line in each approximately averaging the delivery line. Vibrations in this line usually occur when poppet discharge valves are used, and, on account of this, various mechanically moved discharge valves have been devised, the real motive for the designing of which has too evidently been not the promotion of actual efficiency but the securing of *a smoother-looking* card.

The fatal objections to the mechanically or positively moved discharge valve are that the precise times for it to open must vary continually with the delivery pressure, and also to some extent with the speed of the machine, so that if the valve is timed to open for a certain pressure it will open too late for a lower pressure and too early for a higher pressure, causing in each case some loss of power by compressing above the delivery pressure in the one case, and by doing compression work over again when the air blows back in the other. The delivery pressure in the low-pressure cylinder of a two-stage compressor is, of course, constant, being determined by the relative capacities of the high-pressure and the low-pressure cylinders, but this does not relieve the mechanically moved discharge valve as to the more difficult proposition: to have it give free passage for the air right up to the end of the stroke, and yet to have it closed absolutely before the beginning of the return stroke, so that all the air compressed and expelled will be prevented from flowing back.

Of course, it makes all the difference as to what the discharge valve is, its size, its cushion, its spring and all that, which it will not do to go into in detail here. The promptness and efficiency of the poppet valves in closing in this case are sufficiently shown by the re-expansion line, and the slight ultimate clearance.

In computing the volume of free air taken into the low-pressure cylinder and actually delivered, no deduction is called for in this case at the beginning of the compression stroke in either cylinder, on account of deficiency of pressure, as the compression line begins fully up to the atmosphere line. It is to be noted that this compressor not only has the piston inlet, but it has the hurricane piston-inlet valve, a recent and decided improvement upon the original piston-inlet valve. As the piston-inlet valve closes by its own momentum at the end of the stroke, it remains fully open until the actual end of the stroke—the hurricane-valve opening being 12 per cent. of the piston area—and then closes instantly.

The amplitude of the opening provided by the hurricane valve is shown by the near coincidence of the intake line with the atmosphere line of the card. Although we know that the intake pressure in the cylinder during the stroke must be lower than the atmospheric pressure outside, to cause the air to flow in and fill the cylinder, the difference of pressure is here so slight that for most of the length of the stroke the intake line cannot be distinguished from the atmosphere line. At the precise moment of valve closure the momentum of the column of air which has been rushing through the inlet pipe is suddenly checked, but in the stopping of this rush there is a slight increase of pressure at the valve which carries the pressure of the air in the cylinder fully up to or slightly above the atmospheric.

The actual free air delivered, as compared with the total capacity per stroke, is as *a*—*b*: *a*—*d* or as *b*—*a*: *b*—*c*. The total length *a*—*b* of the card being 3.06: 2.94: : 1.00 :0.9607, say, 0.96, which may be called the actual volumetric efficiency of the compressor. The free air actually delivered per double stroke, then, is:

$$21.39 \times 0.96 = 20.53 \text{ cubic feet.}$$

It will be seen that the practice of deducting only the actual clearance space from the full cylinder content per stroke and making that the volumetric measure is not correct in the case of air compression, as the space in which this clearance air re-expands down to initial pressure is also to be included. The actual volumetric efficiency disclosed in this case is really quite high, although not quite as high as some *apparent* efficiencies reported.

We now look at Fig. 3, the cards from the high-pressure cylinder. The diameter of the cylinder being 20¼ inches, the piston-rod 2⅜ inches and the piston-inlet pipe 6 7/16 inches, the net piston areas are:

Rear end (piston inlet), 280.62 square inches.

Forward end (piston-rod), 313.12 square inches.

The full cylinder capacity per double stroke, then, is

$$(280.62 + 313.12) \times 2 \text{ (feet stroke)} \div 144 = 8.24 \text{ cubic feet.}$$

Total length of card, 3.03 inches; total air-intake length, 2.93 inches. Percentage:

$$2.93 \div 3.03 = 0.967.$$

Total intake:

$$8.24 \times 0.967 = 7.97 \text{ cubic feet.}$$

We found that the volume of free air taken into the low-pressure cylinder per double stroke was 20.53 cubic feet. This volume of free air at the intake pressure of the high-pressure cylinder, 23 pounds gage, would be (temperature constant)

$$23 + 14.5 : 14.5 : : 20.53 : 7.93,$$

which is remarkably close to that of the high-pressure intake 7.97. The second cylinder at equal temperature should show slightly the less volume on account of probable slight leakages at stuffing-boxes and elsewhere. The actual result indicates that the intercooler brought the temperature almost, but not quite, down to that at which the air entered the low-pressure cylinder. It will be noticed that I have assumed throughout 14.5 pounds as the atmospheric pressure, the compressor being located high enough above sea level to warrant this.

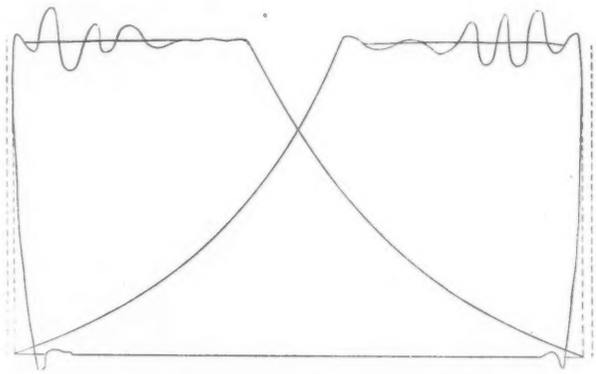


FIG. 2. LOW-PRESSURE CYLINDER

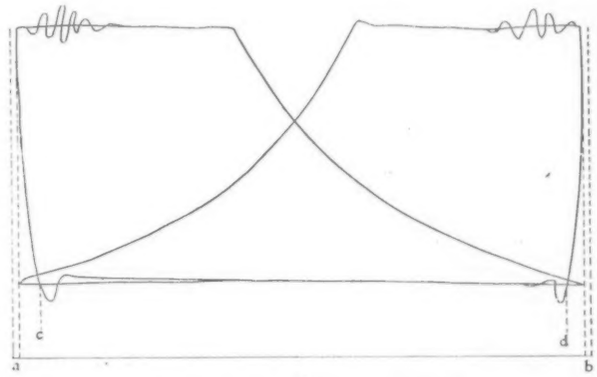


FIG. 3. HIGH-PRESSURE CYLINDER

Fahrenheit and Centigrade temperatures are readily transposed, the one to the other, by means of one of the simple formulas:

$$(1.8 \times \text{Cent. deg.}) + 32 = \text{Fahr. deg.}$$

$$(\text{Fahr. deg.} - 32) \times 0.555 = \text{Cent. deg.}$$

Trials of Engineers in the Early Days—I

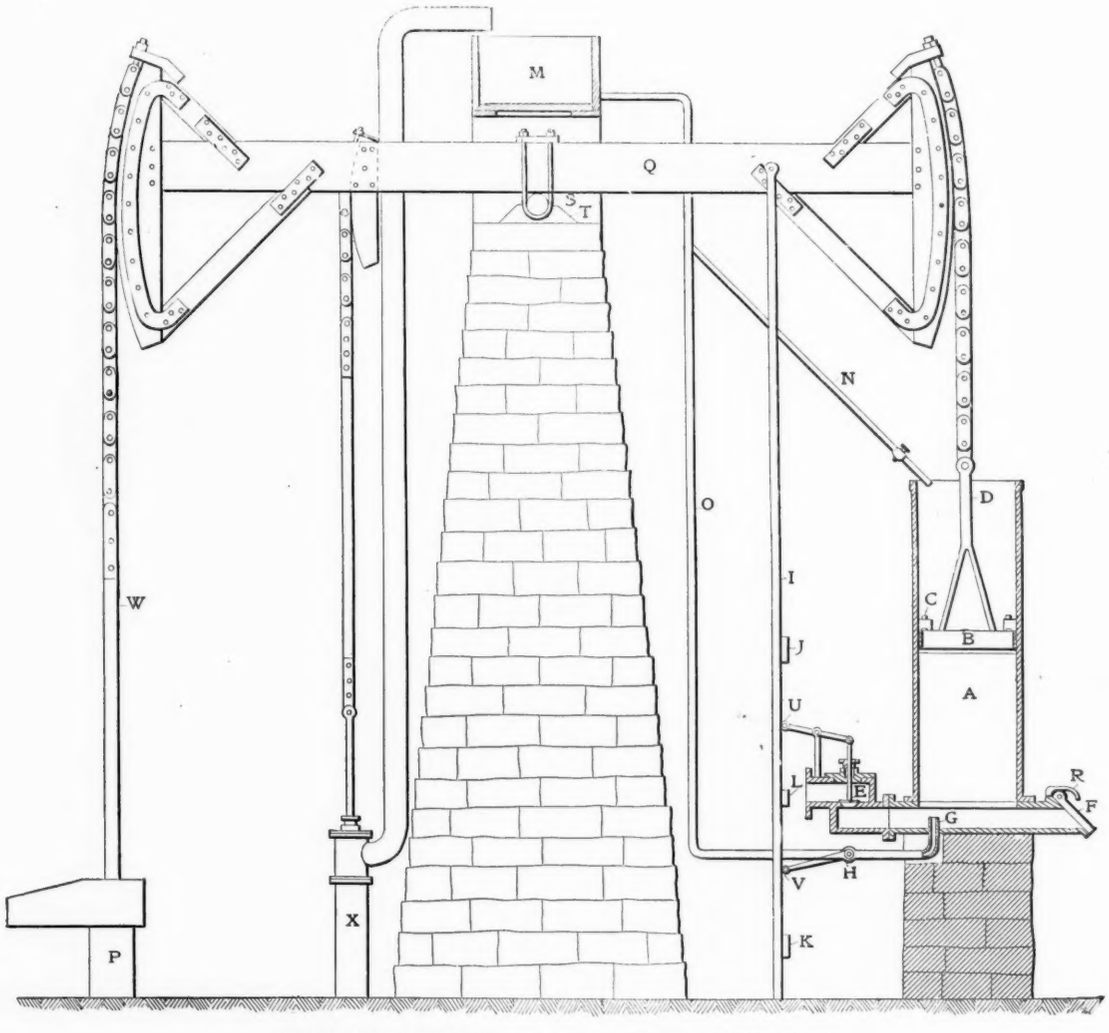
Interesting Results of Research in Rural England for Engine Types of the Time of Newcomen and Watt

EDITED BY ALONZO G. COLLINS

Early one summer a party of engineers made a motor-car trip through the old coal and iron districts of England, in the hope that they might run across some of the early engineering feats of Newcomen or Watt. In Fairbottom valley, near Oldham, they stopped at a wayside inn for lunch, and the landlord informed them that there were ruins of an old coal pit nearby, where it was said there used to be an odd sort of engine for pumping water out of the pit. They procured as guide an old miner, whose father worked in the mine and had often told him about the "fire engine," as he called it. He said that great interest and excitement were caused when the engine was first put at work in the mine.

The tourists had to leave the motor-car at the end of a lane, and make the rest of the way on foot through a tangle of underbrush and briers. When they reached the place they were looking for there were many expressions of disappointment, as there was nothing visible that bore the slightest resemblance to what they expected to find. They were about to retrace their steps, when one of the engineers, a descendant of Richard Reynolds, one of the early ironmasters, declared that he would not give up so easily. The entrances to the mine were almost completely choked with underbrush and fallen débris, but he finally got through and the sight that met his eyes brought forth an exclamation of astonishment which caused his companions to fear that he had met with an accident.

Almost completely hidden by vines and brambles, which in a measure followed their outlines, were found the oak walking-beam, cylinder and pump-rods of a Newcomen pumping engine.



NEWCOMEN PUMPING ENGINE AS PROBABLY CONSTRUCTED

The property belonged to the Earl of Stamford and Warrington, and the steward of the estate was very much surprised that the gentlemen should find anything of interest about a pile of "old junk." He readily allowed them to remove the undergrowth, to permit inspection and make measurements.

The walking-beam is supported on a solid masonry pillar, 7x14 feet at the base. It is of oak braced with iron, is twenty

feet long, twelve inches thick, and fourteen inches deep, with segmental ends to which the piston and pump-rods are attached by chains.

The cylinder is of cast-iron, twenty-seven inches in diameter, and about six feet stroke. Steam entered at the bottom of the cylinder, and after the induction valve had been closed the steam in the cylinder was condensed by a jet of water injected into it, thus forming a vacuum under the piston when the atmospheric pressure on the top of the piston forced it down; on the induction valve being again opened, the condensed steam and injection water were blown out by the entering steam.

Mr. Reynolds, who discovered it, made a drawing of the engine, as he imagined it appeared when first erected. This drawing is reproduced herewith: *A* is the cylinder; *B* the piston; *C* lead weights to hold the piston packing in place; *D* the piston-rod, divided into four forks to prevent canting; *E* the regulator, or induction valve; *F* the snifting valve. In the earlier stages of the steam engine it was the custom to surround this valve with a cistern of water, to prevent air leaks; and the peculiar sound caused by the steam blowing through the water, similar to the sound made by a person sniffing, gave it the name. *G* is the injection nozzle, *H* the injection cock; *I* the plug-rod (so named because the boys who attended to the valves were called plug-boys, i.e., boys who worked the plugs, or cocks); *J, K, L* are tappets projecting from the plug-rod, to which they are adjustably secured, and which, by striking the levers *U* or *V*, open and close the regulator valve and injection cock at the proper times. *M* is the injection water cistern; *N* a pipe to supply water to cover the top of the piston, to prevent air leaking past the packing; *O* injection pipe; *P* mine pump, worked by the rod *W; Q* working beam, or great lever; *R* stop to limit the opening of the snifting valve; *S* the gudgeon or center of working beam; *T* the bearing for the gudgeon, usually formed in a block of granite. The levers *U* and *V* had handles projecting from their sides, by which to operate the valves by hand, in starting the engine.

It must be understood that the pump end of the lever, due to the weight of the pump-rod, is heavier than the steam end. This is made purposely so; as owing to the flexible connection, the steam pressure cannot raise the steam end of the beam; the only purpose of the steam being to assist, by its condensation, in forming a vacuum under the piston. All of the work is done by the pressure of the atmosphere on the upper face of the piston, and as one or two pounds of steam pressure was ample for this purpose, and was as much as was usually carried, the excessive lead given the steam valve by this valve gear was not detrimental. On the up-stroke of the piston it was absolutely necessary to permit the vacuum to be formed before the piston reached its upper limit of travel. This was in order to overcome the momentum of the heavy pump-rods in their descent and bring them to rest before the beam should strike the stops.

Mr. Reynolds tried to imagine how those early engineers contrived to do their work with tools and facilities that must have been crude in the extreme. He determined to prolong his vacation and make a tour of investigation. At Coalbrookdale in Shropshire, it was said a number of Newcomen pumping engines had been built at the iron works owned by the Brothers Darby. He had very little difficulty in locating the successors of the old Darby works, and was highly pleased to find that his ancestor, Richard Reynolds, had been a manager of the old works at about 1760. This at once gave him a standing with the proprietors, who gave him the privilege of searching through a mass of old records and drawings. He was rewarded by finding a journal of one of the old-time workmen, who had spent his spare time setting down his experiences. The introductory pages of the diary were missing. The first whole entry was an account of the making of a steam cylinder, as follows:

"Began this day to scour the bore of a great cylinder of a fire engine for drawing the water from the coal pit at Elphinstone, of a bore twenty-eight inches across, and in length nine feet, the same being cast of brass after much discouragement, and the spoiling of three before, which made us of much doubt if we could ever succeed in a task of such great magnitude; but being, by reason of the extremity to which the proprietors of the pit were at, having to employ more than fifty horses to discharge the water thereof, much urged to persevere, we give great gratitude to Almighty God, who hath brought us through such fiery tribulations to an efficient termination of our arduous labors.

"Having hewed two balks of deal to a suitable shape for the cylinder to lie therein solidly on the earth in the yard, a plumber was procur d to cast a lump of lead of about three hundred weight, which being cast in the cylinder, with a dike of plank and putty either side, did make it of a curve to suit the circumference, by which the scouring was much expedited.

"I then fashioned two iron bars to go around the lead, whereby ropes might be tied, by which the lead might be pulled to and fro by six sturdy and nimble men harnessed to each rope, and by smearing the cylinder with emery and train oil through which the lead was pulled, the circumference of the cylinder on which the lead lay was presently made of a superior smoothness; after which the cylinder being turned a little, and that part made smooth, and so on, until with exquisite pains and much labor the whole circumference was scoured to such a degree of roundness, as to make the longest way across less than the thickness of my little finger greater than the shortest way; which was a matter of much pleasure to me, as being the best that we so far had any knowledge of; but I was busy casting about in my mind for means as to how it might be in future made better, and I reckoned, for one thing, that I would so fashion the iron bars to which the ropes were tied, that they might be laid in the cylinder, and the lead cast on them, and so fasten them firmly."

The greater part of the writing for the next few pages was so faded as to be undecipherable, but it seemed to refer to the construction and fitting of the minor parts of the engine. Further on was the following account of the erection of the engine at the mine:

"The carriage of the engine to the coal pit was a most difficult matter; the wagon on which was the cylinder, being of insufficient strength, was often broken, and in the low places of the road we had often to put twelve sturdy horses to it, but by perseverance and our great ambition it was at last at the end of our journey with no serious mishap.

"We had at first been at some discouragement about the pumps, as we had a very imperfect notion of them; but, succeeding in having the assistance of several admirable and ingenious workmen of Birmingham, we came to the method of making the pump, valves, clacks and buckets in a very superior manner; and they being now arrived, I put some of the workmen to set them in their place in the pit, fashioning strong timbers at the mouth of the pit, and otherwheres, as I judged might be wanted for the proper supporting of it, with the mouth of the pump so high from the ground that the water might find its way to get clear of the pit.

"I now measured with great exactness, from the middle of the pump's bore, away from the pit, half the length of the great lever, that is, thirteen feet and nine inches, and half the thickness of the great chain, that is, three inches, making in all fourteen feet, for the middle of the great pillar, on which to fix the stones for the gudgeons of the great lever, and had a great pit dug to a firm bottom, in which skilled stone masons should build the pillar.

"This I made eighteen feet long in a direction crosswise of the great lever, and nine feet the other way, at the bottom, inclining the sides, so that it should be four feet square at the top and twenty-six feet tall from the ground; it was built in the firmest manner and solid, course by course, with thin lime mortar, of lime that had not been too long slaked; at the top I set stout iron bars, the stones set around them, with screws on the end, by which to fix the gudgeon blocks solidly.

"From the middle of the great pillar,

and in straight measure through the middle of the pump's bore, but on the other side of the pillar from the pump, I measured again half the length of the great lever and half the thickness of the great chain, making fourteen feet as before, for the middle of the cylinder's bore, and set off to each side, lengthwise of the great lever, two stout stone walls on which to build the cylinder beams.

"It had formerly been the custom to place the cylinder over above the boiler; but this had been so troublesome by account of the great straining and jumping of the cylinder, by the great force of the steam, causing the divers joints to leak, and spoiling the stone walls of the furnace, although built ever so solidly, that I made shift to place the boiler a little distance away and to convey the steam to the cylinder by a copper pipe, which by reason of its elasticity would jump with the cylinder and thus, I reckoned, would not so soon spoil the joints.

"I then set the plumbers and coppersmiths to fashion the boiler, with great sheets of copper for the bottom, where the fire was, and similar sheets of lead for the top, which, being preserved from the fire, would not so readily be spoiled.

"The sheets of copper were set with the border of one sheet over the border of another, with hacks or cogs so cut in them as to lock one with the other, and soldered with spelter, like coppers for brewing, until there was one monstrous sheet, which the workmen reckoned large enough for our needs. They then with huge hammers and a mighty din so contrived their blows as to bend the copper to a half circle, like unto half of a globe, with the border turned out to shape a flanch, to which the lead, being similarly shaped, was made firmly solid, with putty between to stop the steam and bars of iron above and below, by screw-bolts. It was a task of greater magnitude than any we had yet undertaken, but with rare skill the ingenious workmen found means for overcoming our difficulties in a most admirable manner.

"Before the boiler could be set in its place it was needful that the cylinder should be got to its place, as otherwise there would be no room for it to pass. This task I desired to take in hand myself, as the other workmen were timorous about how it was to be got up, and a stout heart as well as a nimble body were needed if we were to have no mishaps. So a skilled woodsman fashioned as I directed some round logs of wood, truly round and straight, and others fashioned a sort of frame or cradle of massive timbers in which the cylinder might lie at ease, and the round logs on the ground beneath all, so that it was as if the cylinder was on a wagon, but low down, so that there was but small risk of its overturning; and so, with many sturdy men pulling at ropes in front and others with stout pries pushing from behind, it was at last got between the stone walls on which it was to rest.

"I now must set it on its bottom, upright, which I accomplished by lifting the top end with pries as far as the purchase might go, and, with strong timbers under it, hold it so until a new purchase could be had for the pries, and so on, until it was almost upright, when I fixed ropes to the top whereby some stout men could hold back and thus deter it from overbalancing from its momentum when it came upright, which it came near doing, as the men were caught unawares and were fair dragged some distance; but a shout from me put them on their mettle, and it got no further.

"It was now to be lifted about six feet, to get it to where the flanches on each side could sit on the cylinder beams, and this seemed the most perilous task of all, as it stood so tall and slender, as though a strong wind would suffice to overturn it. I contrived to do it, however, by means of two pries on opposite sides of the cylinder, under the bottom flanch, and four ropes from the top, with some sturdy men to pull four ways to keep it upright, whereby lifting on the pries steadily and sedately, as far as the purchase would go, and by building up with planks laid evenly under the flanch, between the inner ends of the pries, the cylinder was held up until a new purchase was made for the pries, and so on; and I was much pleased to have it go so smoothly.

"It was fairly the easiest part of it, and it was done so much speedier than I deemed possible, that I was constrained to let the men lie by for the rest of the day, as a token of good will for their nimbleness.

"The joint between the cylinder and the bottom I made firm with lead, beaten thin where the space was slim, so to match the unevenness of the flanches, and after the bolts were screwed firm, by beating the flanches with hammers and warming with a small fire, to soften the lead, I at last got it so that it was stanch with the cylinder filled with water; the other joints in the pipes, and so forth, I made firm with putty.

"I had now to contrive the great lever, which must be twenty-seven feet and six inches long; the gudgeon was of cast-iron, twenty inches broad and five inches thick, where it was fixed in the wood, and I reckoned to make the lever of two balks of sound fir, twelve inches square, one above and one below the gudgeon, cogged on the gudgeon half an inch each side, and they would thus be spread four inches at the middle; outside of these I put two sound fir timbers, six inches deep and twelve inches broad, each above and below, which measures the lever four feet and four inches at the middle, and pulling them together by twelve iron bolts one inch thick, so it was four inches less at the ends.

"I built a curve or arch head on each end, of a length to suffice for nine feet of chain equally above and below, the curve being part of a circle of the same diameter as the length of the lever, so that the piston-rod in the cylinder at one end and the pump-rod at the other end of the lever should be constrained to move up and down truly perpendicular. The curves I made very strong and sturdy, of fir, with iron bars on each side, hammered to match the curve and bent back above and below to meet the lever, and bolted with screw-bolts to make it very firm.

"The pump was for a stroke seven feet and six inches long, and I contrived the great chains so that the piston should rise to within six inches of the top of the cylinder and sink to within six inches of the bottom, which, with the stroke, seven feet and six inches, and the thickness of the piston, six inches, made the nine feet in length of the cylinder, and fashioned two stout preventer posts, one on each side of the great pillar and squarely under the great lever, so that the great lever will be stopped by them should the engine be constrained to make too long a stroke.

"The pump-rods I fashioned of sound fir four inches square, the ends jointed with iron bars and bolts, firm and sturdy.

"I now perceived where I had been at fault through my ignorance, in that I had supplied no place for the cistern for the cooling water, which must be on a tall place to cause the water to enter the water case speedily; but I contrived some timbers above the great lever, on the top of the great pillar, which did answer very well, but I had a very imperfect knowledge as to how I should get the water to it, and cast about to find some brook to back up by a dam, but found nothing to my liking, until one of the workmen contrived a little pump with clacks and bucket, which should derive its water from the top of the great pump.

"I fashioned a small curve on the side of the great lever, half-way between the end and the middle, whereby with chain and pump-rod to give motion to it, and a lead pipe to convey the water to the cistern, also a lead pipe from the bottom of the cistern to the water case, and a branch pipe to discharge water on top of the piston to stop air from going in the cylinder, with a cock in each to regulate the water."

A new hoisting rope should be oiled within two or three days of its first being put to work. The oiling should be repeated at least once a month for the first six months, and then every six weeks. In a dry, down-cast shaft this may be considered needlessly frequent, but in an up-cast or wet shaft the oiling may be required more often. The process of oiling affords an excellent opportunity for examining the condition of the rope.—*Mechanical World.*

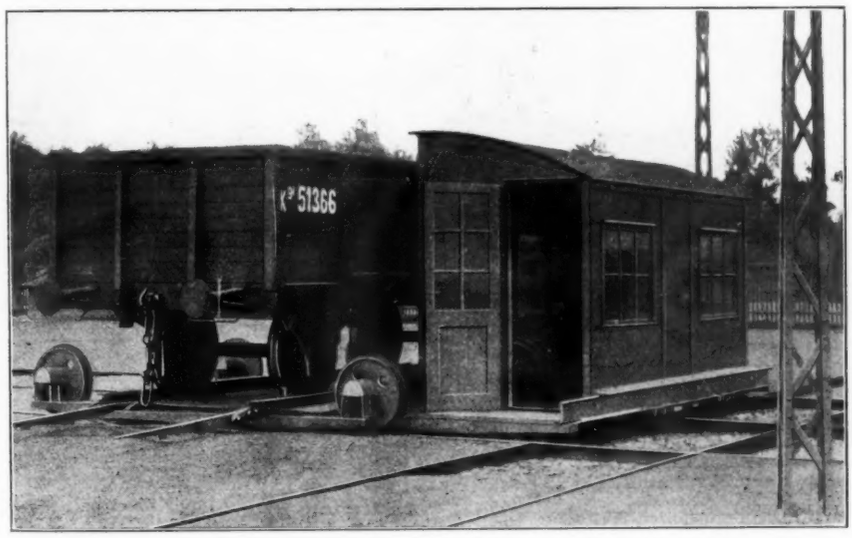
Coal Storage and Coal Conveying Systems

Some of the Modern Methods Employed in Handling Fuel for Large Power Stations at Home and Abroad

BY FRANZ KOESTER

The fuel question is one of the live subjects before the power-plant manager; the plant cannot run without fuel, and the cost of this article is the largest single item in the operating account. The cost of the coal per ton, as taken from the bill, is not the total cost of that amount of coal, for this only includes its delivery in cars or boats at the power plant, where it has to be unloaded, conveyed to the fire-room, placed on the grate, and the resulting ashes must be removed from the building and disposed of in a suitable manner.

The quantity of coal required is from ten to twenty-five times the weight of the resulting ashes and for this reason proximity to a point where coal can be secured with the minimum amount of handling is of more importance than ability to dispose of the ashes alongside of the station. In any event it is desirable to avoid carting either coal or ashes, as such procedure means additional handling and its attendant expense.



TRANSFER TABLE OF THE COAL-HANDLING SYSTEM AT THE VIENNA TWIN MUNICIPAL PLANT

As plants increase in size the importance of a reserve supply of coal becomes more and more evident. For the small plant whose consumption is limited to from five to ten tons of coal per day, one or two carloads is often considered a big supply on hand, while in the plant whose daily requirements run from six hundred to a thousand tons of coal, a large reserve is necessary, and, as will be seen later, the New York Edison Company has found it advisable to put in a storage plant, in addition to the reserve carried in the bunkers of the power plants, of 150,000 tons capacity.

The uncertainty of the coal supply is due to many causes. Strikes in the mines of from a few days to a number of months duration have often occurred; in some cases these labor troubles have been sufficient in magnitude to practically stop the mining of coal. All transportation service is subject to interruptions of a more or less serious character from storms, floods and, in some cases, strikes; these last, however, have not been frequent in late years, but a number of railroads have had considerable trouble with freight handlers in some localities. Where railroads are depended upon for coal supplies, the first signs of labor troubles at the mines is the signal for them to seize all the coal in transit on their rails and until their stores are filled to the limit there is little chance for an outsider to secure any quantity of coal.

As, except in a few instances, it is impossible for the coal consumer to have absolute control of his supply from the mine to the fire-room, it is absolutely necessary to adopt such measures as will protect his interest during all contingencies, and the only method by which continuity of plant operation can be insured under all circumstances is by the establishment of a coal-storage plant of sufficient size to carry the plant over any stoppage in the receipt of coal.

In some plants the coal bunkers alone are depended upon as a reserve, while in other cases they have been reinforced by storage plants designed for the purpose. A fuel-storage plant of any magnitude involves a considerable investment in the machinery required, and in addition a large amount of money is tied up in the coal pile. The interest on this money and the operating costs of the plant add continually to the fuel bill, but to offset this the insurance value of the plant in times of trouble is beyond computation.

In most coal-storage plants the coal piles are exposed to the weather, and for practical reasons it is essential that a portion, if not all of the daily supply, be handled, in order to keep the machinery in shape, and for other reasons, as will appear further on. Coal stored in the open, or under cover, loses a portion of its heating value, this loss, in fact, being due to slow chemical changes which start as soon as it is exposed in the breast at the mine, and continue until it is burned. Such losses are greatest with bituminous coals, as the anthracite coals depreciate less rapidly. They are greater, too, in warm weather, and in the tropics, and less in cold weather. These losses are due in part to the coal giving up its occluded gases, which, in some cases have a considerable thermal value. The extent of this depreciation is slight when the coal is fresh, but increases rapidly the longer the coal remains in the pile.

Causes of Spontaneous Ignition

Coal-pile fires are not of infrequent occurrence, although it is very rarely that such fires are chronicled. Their cause is somewhat obscure, but they occur more frequently in bituminous coal, which contains considerable sulphur in the form of iron pyrites, though all coals of a friable nature are liable to spontaneous ignition. There is considerable difference of opinion in regard to the causes of these fires. Moisture is by some considered to retard such ignition, but wet coal in compact masses has been known to catch fire spontaneously. Ventilation sufficient to evaporate the water and keep the heat of the pile down is probably the best method of overcoming such trouble, but restricted or insufficient ventilation is likely to increase the trouble.

Spontaneous ignition is just as liable to

occur in coal stored under cover as when open piles are used, but wet coal containing considerable pyrites will ignite if stored long enough to allow heating. Such fires are discoverable by the odor of the gases, difficulty in breathing at such portions of the pile, and by the heating of the coal. This last can be discovered by driving down pipes, through which thermometers can be lowered and the temperature taken, or in winter by the fact that snow will not remain on the pile, but melts away. It is impossible to fight such fires by water, unless the whole pile can be flooded and the entire air supply cut off in this way, for the reason that the fire cokes the coal adjacent to it previous to its burning, forming a roof which would shed most of the water, provided the water is not evaporated before it gets this far down in the pile, as is usually the case.

In some cases pointed perforated iron pipes can be driven down in the pile, if it is not too deep, and water forced to the fire through them, but the most complete method is to remove the coal over the fire and then attack it.

Another trouble with exposed coal piles and railroad shipments occurs in winter when the coal often arrives frozen solid in the car, and the unloading tracks must be equipped with steam pipes and outlets for thawing purposes. The coal piles form a frozen crust which must often be broken up with dynamite in order to get out the daily requirements.

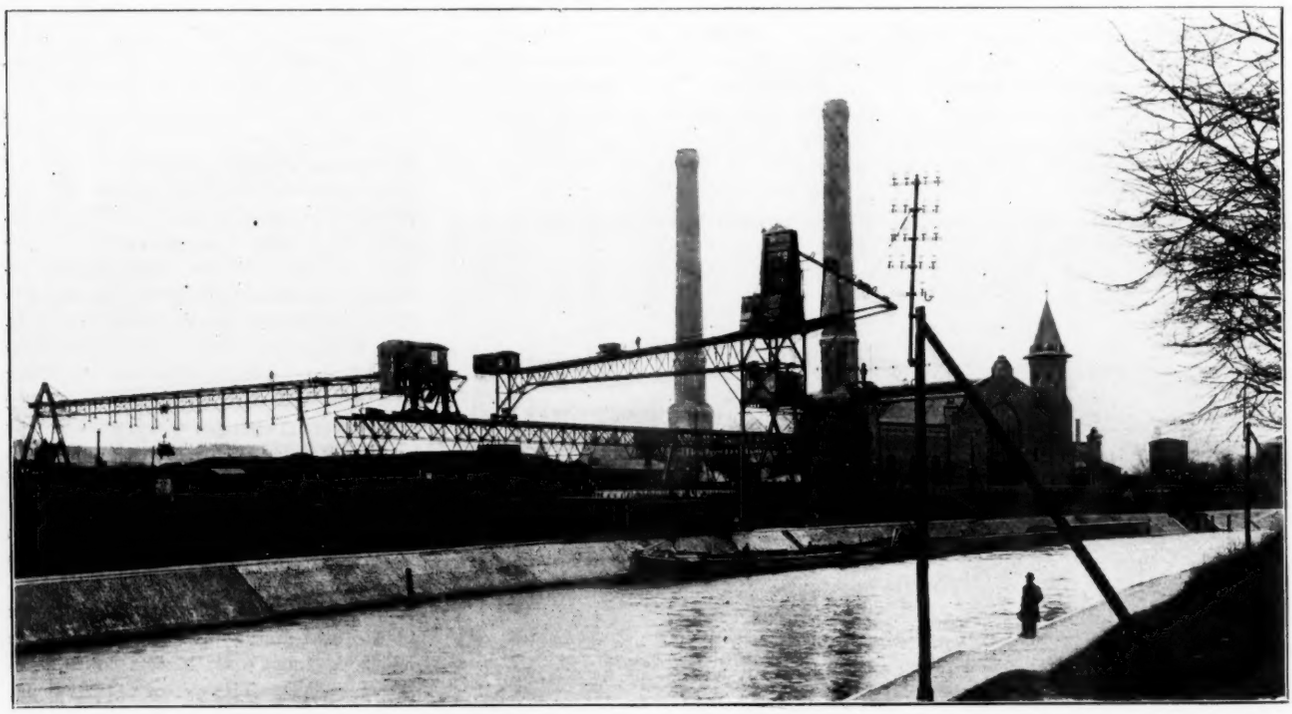
From the foregoing it can be seen that the storage question is not a simple one, and must be handled with a full understanding of the many trials and tribulations that are likely to ensue.

The general arrangement and the plan required for coal storage depend upon local conditions, and the concerns which build coal-handling machinery are prepared to submit designs for plants of any desired capacity, when they are furnished with ground plans showing the area to be covered and its contour. Some plants require level ground or expensive grading, while in other cases conveyers or other machinery can be installed with but a slight amount of grading. The character of the plant will depend in part upon the method by which the coal is to be received and delivered; in some places coal is received by cars alone, in others boat coal only is used, and occasionally both methods are available. Not infrequently the coal is transported to the power plant by conveyers, and in one notable instance it has to be reloaded on barges to reach the power house.

In Continental plants the coal-storage plant is generally located alongside the boiler house, and is usually covered or roofed in, and the coal is brought into the firing room by small cars; but in a few of the recent plants American practice has been followed, overhead bunkers and conveyers being introduced and the coal spouted down to the boilers, large exposed piles containing the reserve coal being located at the side of the boiler-room.

The coal-storage plant of the Edison Electric Illuminating Company, of Boston, Mass., presents a very interesting method of treating this subject. At this plant the fuel is delivered in large barges, carrying over 1000 tons each, and two channels have been dredged out and wharves built, on both sides of which the barges can be moored. On the wharf are mounted two unloading towers, a small one installed at the time the original power station was built, and a second tower added at the time the turbine station was built.* There is room for an additional coal tower on the pier, which will be erected when the extension of the station necessitates such addition to the handling capacity. The first tower requires two operators, is equipped with a one-ton bucket, making one trip per minute. The second tower requires but one operator, and can unload from either side of the pier; it is equipped with a 1½-ton grab-bucket, which can make three trips per minute.

The coal is transported to the coal-storage yard by Robins belt conveyers and is deposited along one side of the yard from the conveyer trestle. The storage yard is covered by a bridge tramway, which distributes the coal over the yard or redeposits it on the conveyer, which can be reversed for transporting the coal to the bunkers. The storage capacity of this plant is about 100,000 tons.



COAL-HANDLING SYSTEM AT THE "OBERSPREE" POWER PLANT, BERLIN

Edison Company's Shadyside Plant

The New York Edison Company has a storage plant at Shadyside, N. J., where facilities are provided for storing 50,000 tons of bituminous coal and 100,000 tons of anthracite. This arrangement is peculiar, owing to the fact that the storage yard is located a considerable distance from any power plant of this company, for the reason that New York City real estate is too expensive to pile coal on in large amounts, and, likewise, because railroad connections could not be made to

*See POWER for July, 1905, page 389.

any of the plants. The location chosen for the Shadyside storage plant permits of connection with several railroads and arrangements are made by which anthracite can be received either by cars or by boats. The bituminous coal is received entirely by boats, and both kinds of coal must be reloaded into boats for transportation to the stations of the company in New York City. This plant was designed by the Dodge Coal Storage Company, of Philadelphia, Penn.

The bituminous-coal pile is covered by a traversing bridge equipped with a two-ton clam-shell bucket, which unloads from boats to the pile or from the pile to boats, the front tower in which the operator is stationed being provided with suitable loading chutes.

Anthracite can be unloaded from boats by a steeple tower with a 1½-ton grab-bucket, which delivers to a reversible belt conveyer, by which the coal is transported to a hopper and passes to the trimming bridges spanning the piles. Railroad coal is dumped into a track hopper, from which it passes to the trimmers. The anthracite piles are of the well-known Dodge circular type and a reloading sweep traveling on semi-circular tracks is used for taking the coal from the piles to a hopper, through which it reaches the belt conveyer to the steeple tower, where it connects with a chute for loading boats. Five-ton weighing hoppers, in duplicate, are provided in the towers, by which all coal can be weighed up on its receipt and shipment.

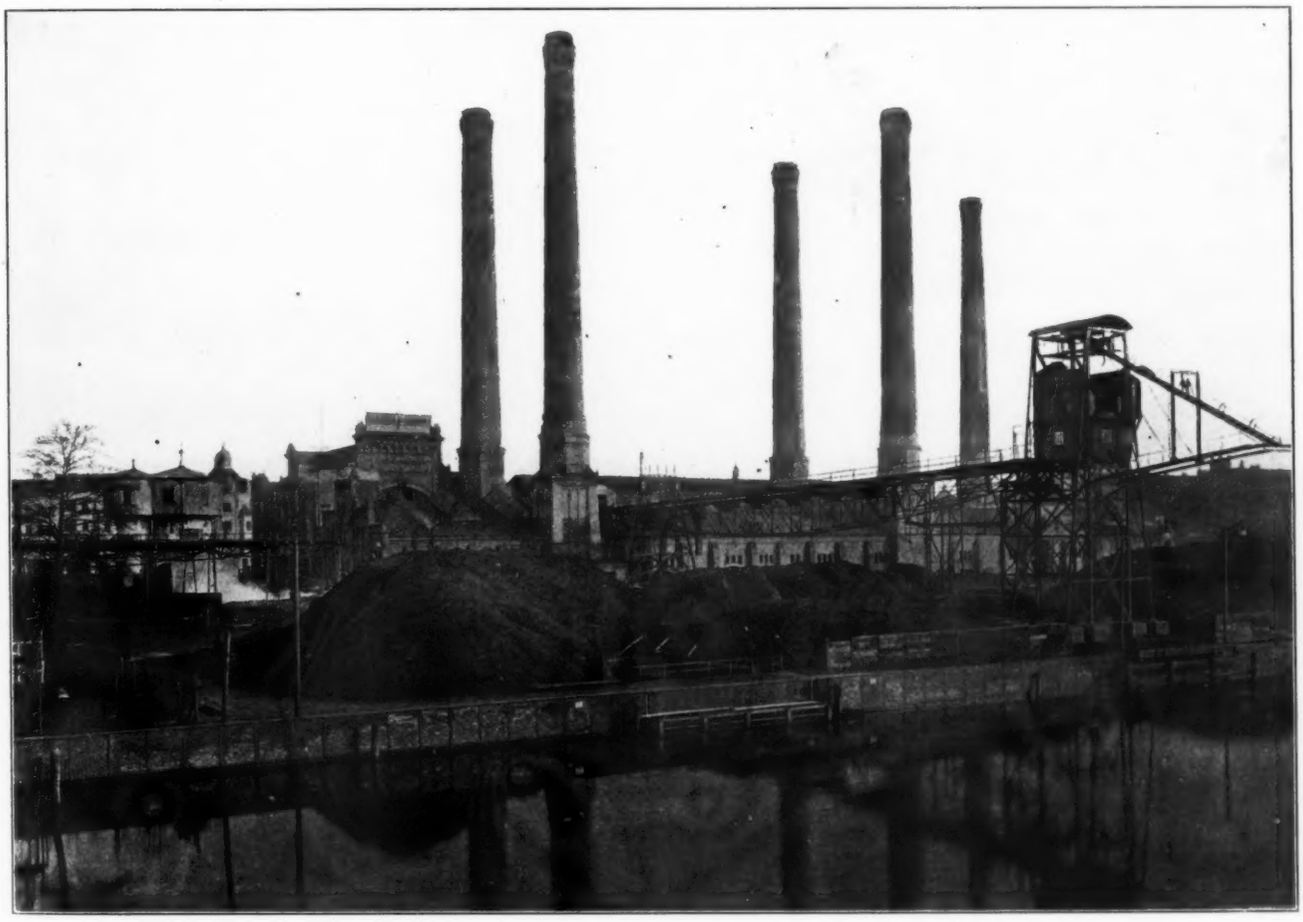
At the plant of the Metropolitan Electric Supply Company, London, England, a coal hopper has been built below the track on which the coal is received. Over this hopper is placed a Bennis rotary tipping device in which the coal cars, after being clamped, can be turned completely over, the coal falling into the hopper, from which conveyers transport it to the bunker in the boiler-room.

On the continent of Europe a great deal of hand labor is employed in the coaling plants. For instance, at the Barmbeck plant, Hamburg, Germany, the coal is received in barges, and is shoveled into buckets, which have a trolley attached to the bail and are lifted from the barge by a traveling revolving crane or "whirley," which places the trolley on an inclined track, down which it runs into the storeroom, dumping at a determined point and passing on around the loop, and back to a point where it can be reached by the crane. There are six of these loops alongside of each other, and the cranes can take from or place buckets on any one of them. The storage capacity is about 6000 tons. The coal is loaded by hand in three-wheeled cars, by which it is taken to the fire-room, and is shoveled from the car into the furnace.

A Novel System in Vienna

A novel system involving much hand labor, both the receipt of coal and the firing being attended with additional expense, is exemplified by the Vienna Twin Municipal plant in Austria. The coal is brought in by cars on a railroad siding running between the two plants. The cars are passed over a scale and are weighed; they are then placed on a transfer table, as shown in one of the accompanying illustrations, by means of which they are transported laterally to either of the two plants. From the transfer table the cars pass to tracks leading to elevators, by which they are raised to tracks passing over the coal bins, twenty-five feet above the bottom of the bins, which are on a level with the boiler floor. The coal is deposited in these bins according to its quality, and from them is loaded into small cars by hand for transfer to the fire-room, and is shoveled from these cars into the furnaces.

These are a few of the European coal-handling systems which would be seldom found in connection with American power plants, when the coal is conveyed by the chain-bucket system, and in some in-



COAL-STORAGE SYSTEM AT THE BERLIN ELEKTRICITAETS WERKE

stances by belt conveyers. This is, from the writer's viewpoint, largely due to the layout and construction of the buildings. The reason, to a certain extent, is that in the European plants, where the cost of land is comparatively small, the buildings are low and the coal bunkers are at the side, while in America, where the land is much more expensive, the buildings are designed higher with the bunkers on top.

It must, however, be stated in connection with recently constructed prominent plants, such as the Chelsea power plant of London, the St. Denis plant, of Paris,* and the power plant of the Berlin subway and elevated road, that in these instances belt Europe than in America, as in the former the capacity of the cars rarely exceeds twenty-five tons, while in this country cars up to forty and fifty tons capacity are used. Here, as well as abroad, the tendency toward conveying coal in large quantities is very noticeable, as, for instance, in the case of the Mead-Morrison coal-handling system at the Gould street power house, Baltimore, where the coal is hoisted by a 25-horse-power steam engine up an inclined structure in buckets of one-ton capacity, and dumped into a hopper in the tower, after it has passed the crusher at the base, operated by a 15-horse-power motor. From here it is is transferred to a second belt conveyer. It is then raised by a series of belt conveyers to the top of the bunker, 100 feet above the boiler-room floor, during the course of which journey the coal is transferred four or five times, as there are two longitudinal-belt conveyer systems at the top of the bunker. It will be seen that each conveyer must be operated by a separate motor, on account of which the liability of breakdown is materially increased, and of course the failure of one piece of apparatus means the disabling of the entire system.

In studying this installation, the question immediately arises as to why a ver-

COAL-CAR TURNING DEVICE AT METROPOLITAN ELECTRIC STATION, LONBON

and bucket conveyers have been employed, and without doubt in the design of future power plants in Europe the overhead bunker will be much more employed, whether mechanical stokers are installed, or whether the old method of hand firing is adhered to, which will naturally cause a more common use of the conveyer system. The question still remains, however, as to whether they will convey the coal in small quantities or in greater bulk, similar to that at the above mentioned Vienna plant, where the whole coal car is hoisted and dumped into the bunkers.

This may be done far more readily in dropped into cars operated by cables from a 10-horse-power motor.

Another large system worthy of attention is that at the Long Island City Railway power plant, where the coal is hoisted in grab-buckets of 1½ tons capacity to a hight of about 160 feet, when it is dumped into a hopper supplying the crusher, from where it falls into the cars of a cable road which brings the coal to the top of the bunkers in the power house.

In some plants, noticeably that of the Interborough Rapid Transit Company, the coal handling is accomplished entirely by a belt conveyer, i.e., after the coal passes the crusher at the coaling tower it is brought on belt conveyers to the side of the power house, during which course it

tical-bucket system, or a system similar to that at the Long Island City plant, was not employed. As this belt-conveyer system occupies the entire end wall of the boiler-room with the crossing of belts, the vertical system would have saved not only a large amount of space, but would have also materially decreased the operating cost. Besides this, it is claimed by various authorities that the entire coal-handling plant of the Long Island City station, which is undoubtedly one of the most ideally constructed coal-handling systems in modern practice, could be installed for a small fraction of the cost of the above described cross-belt conveyer system.

Of course the writer must admit that the belt conveyer may have some advan-

*POWER, February, 1907. The Largest Steam-turbine Plant in Europe. By Franz Koester.

tage, especially on small elevations. Where, however, side runs are necessary enormous space is required, another example of which is the new Boston Edison power station. Another disadvantage of the belt-conveyer system is that the belt can hardly handle the ashes which may be wet, or more usually red hot, as they come out of the ash-hopper of the boilers. This is a serious point on account of the common practice of utilizing the same conveying apparatus for the handling of both coal and ashes.

Overhead Bunkers

Overhead bunkers are usually found when the plant is located on expensive real estate. Such plants are usually far above the average in bunker capacity, which ranges from five to thirty-five tons per lineal foot. These bunkers are composed of a steel framing supporting concrete or reinforced-concrete flooring, upon which the coal lies. The bottom of the bunker slopes usually at an angle of 45 degrees, and forms a series of pockets with inverted V-shaped division slopes, the coal chutes being attached to the bottom of these pockets. For a double row of boilers the cross-section of bunker is like a W, the outside sometimes being carried up vertically to give added depth when the spread has been carried to the rear boiler columns.

In some bunkers the floor slopes are made very flat, but it is not advisable to use a slope of less than the angle of repose of coal; say, 27 degrees for anthracite and 35 degrees for bituminous coal; and under ordinary conditions such bunkers will be self-clearing. Convenience in framing, however, makes the 45-degree slope more desirable, and gives an added factor to the self-clearing properties of the bunker, though under unfavorable conditions, particularly after bunker fires have occurred, it may be necessary to send men into the bunker. The angle of repose mentioned is the natural slope taken by the coal, and it may be of use in estimating the capacity of coal piles and bunkers when the coal extends above the top in the pile.

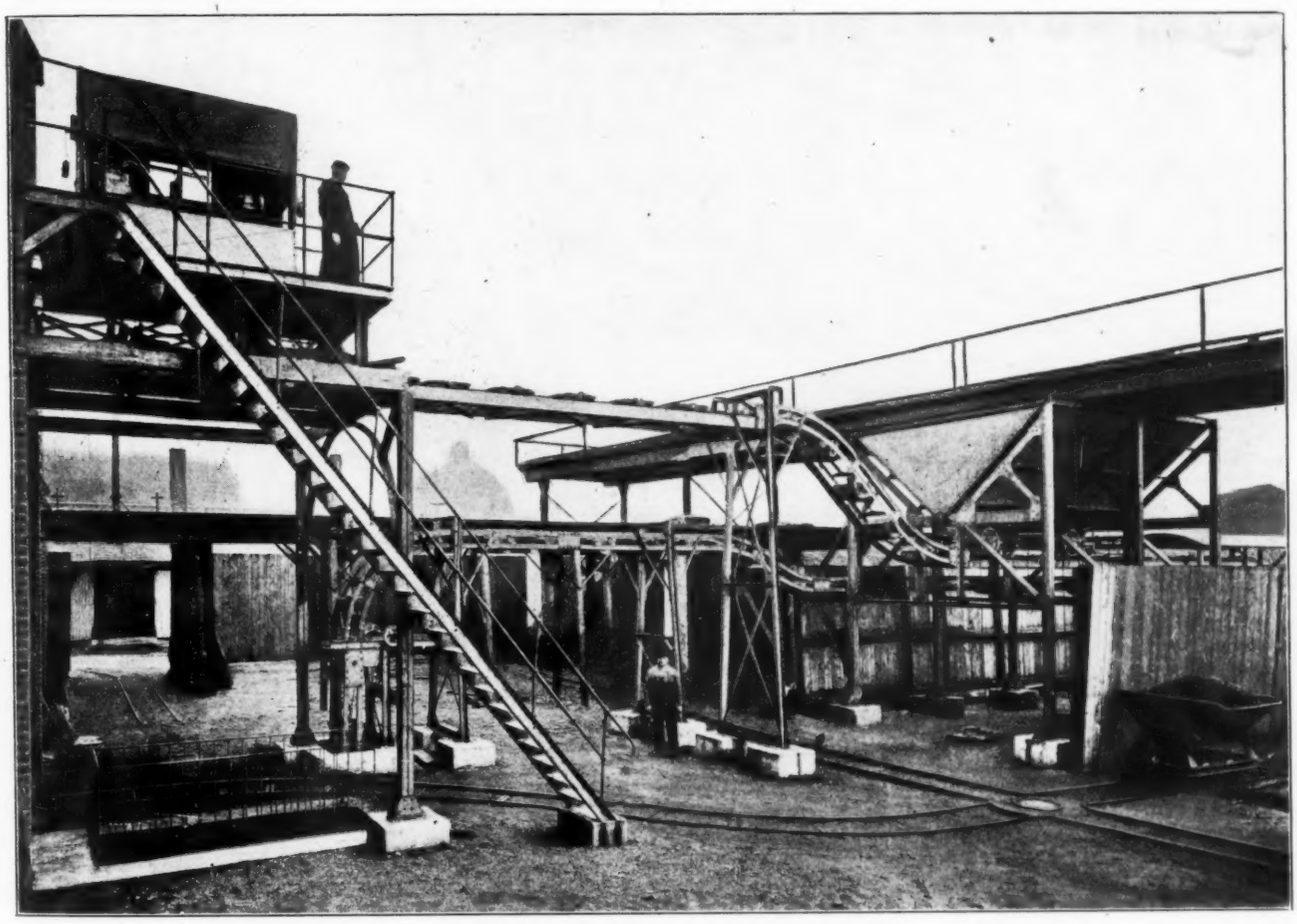
In a few plants the mistake has been made of using such a flat-bottomed bunker that hand labor is necessary to shovel the coal over the spouts. Such bunkers are dangerous and frequent fires occur. Another type is the suspended bunker of plate-steel construction, which is rarely lined with concrete. Plate-steel bunkers usually run from five to ten tons per lineal foot.

The use of bunkers is essential to proper utilization of mechanical stokers in which the feed-hoppers are usually too high to permit of the coal being fed to them conveniently by hand. In existing plants, when mechanical stokers are installed, a small coal pocket suspended from the front boiler columns or the roof trusses is sometimes provided for each boiler, and in other cases a continuous steel pocket is put in.

The openings in the bottom of the bunkers may be provided with cut-off gates; with steel-plate construction the gate frames can be secured to it, but in concrete bunkers it is desirable to provide a casting at the throat of the pocket, which will resist the cutting action of the moving coal, and furnish a convenient point of attachment for chutes or gates.

In some cases the chutes are bolted directly to the bunker casting with a cut-off gate in it, while in other cases a small receiving hopper is suspended from the bunker or the steel frame of the building, or a weighing hopper may be placed at this point. The coal spouts lead down from these hoppers. Sloping spouts are preferable to those in a vertical position, as the coal has a tendency to hang in vertical pipes and considerable batting is often necessary to start it running. Sometimes the weight of coal in these spouts is found by experiment, and it is endeavored to keep track of coal consumption by counting the number of times the spout is filled.

The spouts are usually of cast-iron of from twelve to eighteen inches in diameter. Square sections have been occasion-



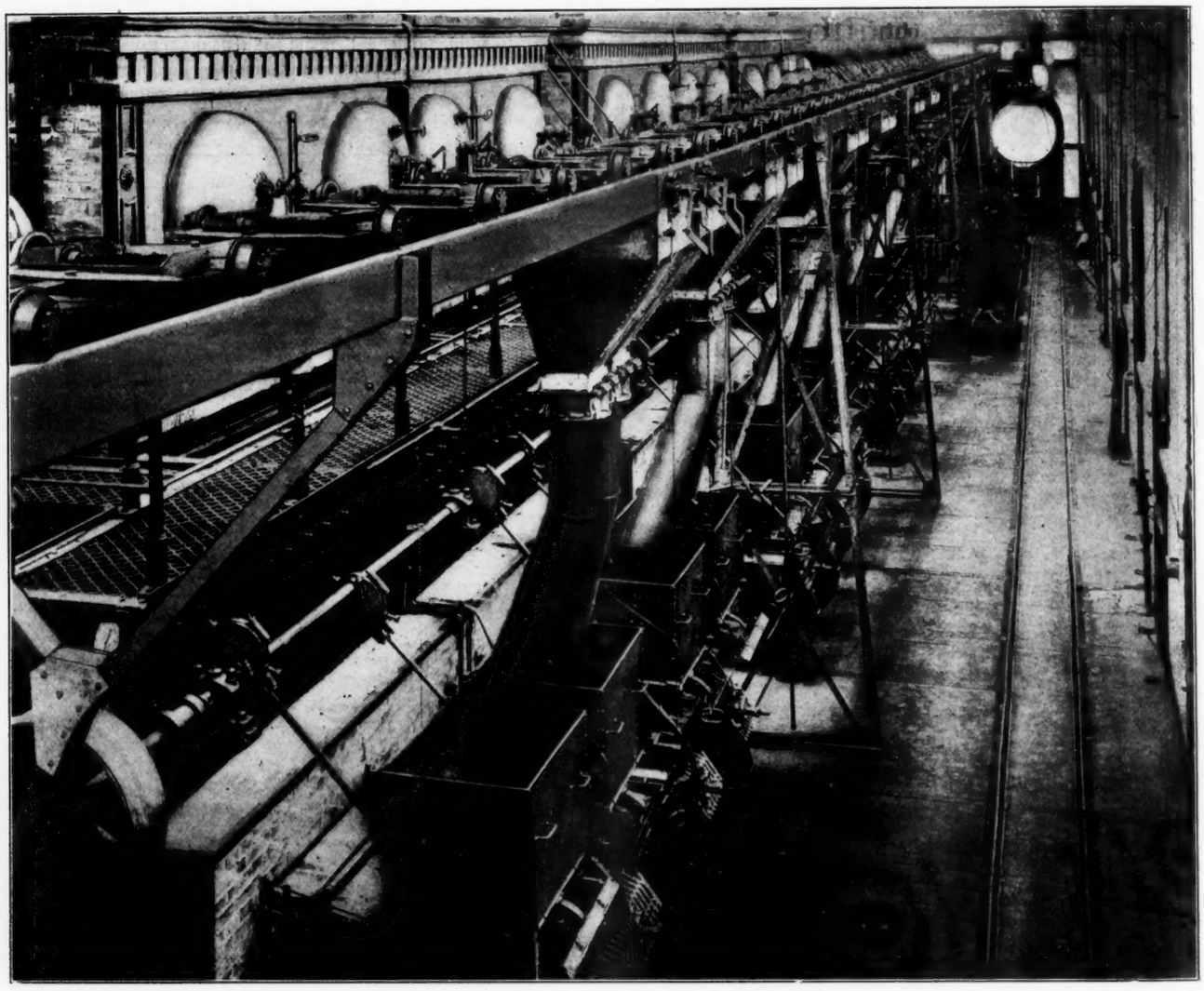
"SCHENCK" COAL-HANDLING SYSTEM AT FRANKFORT-ON-THE-MAIN, GERMANY

ally used, it being claimed that the coal does not hang in such spouts as badly as it does in those of circular section. When over-feed stokers are used a spreader apron is required on the lower part of the spout, or else a hinge is arranged so that the coal can be spread by swinging the spout laterally. It is also necessary to hinge the spouts so that they can be drawn back out of the way when repairs are to be made to the boiler.

In some plants a traveling hopper and chute is installed, having a capacity depending upon the design of the plant, in which the coal is received from the bunkers and distributed to the boilers as required. This hopper can be arranged with scales by which accurate account can be kept of the fuel consumption.

Fires are not infrequent in overhead bunkers, owing to their being exposed below to the heat of the boiler-room, and also because wet coal is frequently stored. Provision has been made in a number of plants by overhead water lines for fighting such fires, but experience proves that water is a poor method of combating them, as they are usually in the lower part of the bunker, and water reaches everything but the fire. Steam can be injected into the bottom of the bunkers through the gates, or, as these fires may occur with considerable frequency, it might be desirable to install a special pipe system to smother them.

The best method, however, to control a bunker fire in the shortest possible time is to empty the particular bunker through the spouts and attack the fire at the boiler-room floor. Where the coal-handling facilities are such as to allow of distributing the coal throughout a number of boilers it might be well to immediately convey the coal which is on fire to the boilers and have it burned up as rapidly as possible. While this method may seem radical, and may mean a considerable loss in coal, it is, in the writer's opinion, the best method of procedure, as the loss entailed would be but a very small item compared to what it would be were the fire allowed to gain a headway and spread over the entire supply of fuel.



INTERIOR OF BOILER HOUSE, SHOWING COAL-HANDLING SYSTEM IN A POWER PLANT AT FRANKFORT-ON-THE-MAIN

Another Lap-seam Explosion

At 6:30 on the morning of Tuesday, January 29, a 42-inch boiler 14 feet in length, with one-quarter-inch head and plate, used at the Maud S Wind and Pump Company plant, at Lansing, Mich., exploded by the failure of the longitudinal lap-seam joint. It was of iron and twenty five years old.

In an article entitled "The Cost of a Golden Age," presented by A. S. Fitch, in a recent issue of the *American Machinist*, it is shown that nearly fifteen billions have been expended within the past sixty years to protect inventions by patents. Adding interest at the rate of six per cent. for thirty years, the amount is brought up to almost forty-five billions, a sum exceeding the national debt of all the countries of the world. The estimated cost includes not only the Government fees, but those of solicitors, development expenses, the cost of litigation, and promoters' charges.

Practical Data from Modern Gas Engines

Fuel, Water and Oil Consumption of Some Representative Types and Their Weight and Space Requirements

BY L. L. BREWER

The writer has found in his experience as a gas-engine salesman that the customers in this particular line are, as a class, about the most inquisitive persons one can meet. The particular merits of any one machine are of interest only after the customer is convinced that the gas engine is to be taken seriously, and that it promises such a saving that he can afford to take a chance that it will not develop some trait that he has forgotten to inquire about.

Every consideration which can possibly affect the cost of operation of a gas-power plant is a matter of careful inquiry on the part of the customer, who usually appears to take the salesman's statements with a grain or several grains of salt, which does not seem to be true when selling steam plants.

It would seem, therefore, from the foregoing that some information gathered from engines in successful operation should be useful to both parties.

Such unbiased information is embodied in the report of a committee of engineers who were sent to Europe to learn these facts, and the results of their investigation have come to the notice of the writer.

The knowledge most sought by buyers seems to be that in reference to the weights and floor space occupied, the consumption of gas, water and lubricants, and the efficiencies to be expected in everyday operation. Such information is given below for Europe, where gas engines have been in operation much longer than in this country.

Weight and Floor Space

These data can best be shown in tabular form, and in the accompanying Table 1 such reduction has been made as to enable the reader to make comparisons on a common basis.

While the revolutions per minute of the Cockerill engines named are less than the others, their longer strokes bring the piston speed up to about 825 feet per minute, which is considered about the maximum for good practice.

The references in the table are:

1. Brake horse-power.
2. Revolutions per minute.
3. Builders.
4. Number of cylinders.
5. and 6. System of operation:
   4. Four-cycle;
   2. Two-cycle;
   s. Single-cycle;
   d. Double-acting.
7. Arrangement.
   s. c. Single cylinder;
   tw. Twin;
   td. Tandem;
   d. tw. Double twin;
   d. td. Double tandem.
8. Weight in pounds of engine without fly-wheel.
9. Weight in pounds of fly-wheel sufficient for driving blowing cylinders, pumps, etc.
10. Weight in pounds of fly-wheel sufficient for driving direct-current generators.
11. Weight in pounds of engine with fly-wheel (9) per brake horse-power.
12. Weight in pounds of engine with fly-wheel (10) per brake horse-power.
13. Floor space occupied in square feet per brake horse-power.

weights given for the four-cycle four-cylinder and the two-cycle two-cylinder engines are ample for the parallel running of alternating-current generators; for instance, the single-cylinder Cockerill engine of 600 brake horse-power would require a fly-wheel of over 400,000 pounds weight, and the 1200-horse-power tandem engine one of 275,000 pounds weight, which would be excessive.

Consumption of Gas, Water and Lubricants

The gas consumption of gas engines of equal quality is higher with waste gases than with the illuminating gas, inasmuch as the richer gas permits a more complete combustion than does the poorer, and this results in a considerable difference in the thoroughness of the utilization of the fuel. Experience has, however, taught the means for overcoming these

TABLE I. WEIGHTS AND FLOOR SPACE, CONSUMPTION OF GAS, WATER AND LUBRICANTS, AND EFFICIENCIES LOOKED FOR

1	2	3	4	5	6	7	8	9	10	11	12	13
Brake Horse Power.	Rev. per Minute.	Builder.	No. Cylinders.	Strokes per Cycle.	Single or Double Acting.	Cyl. Arrangement.	Weight of Engine Without Fly-wheel, Lbs.	Weight of Fly-wheel for Blowing and Pumping Service, Lbs.	Weight of Fly-wheel for Dynamo Driving.	Weight of Engine and Fly-wheel (9) Per B. H. P.	Weight of Engine and Fly-wheel (10) Per B. H. P.	Sq. ft. Floor Space Per Brake H. P.
100	150	Cockerill.....	1	4	s.	sc.	45,000	9,000	21,100	540	661	2.05
200	105	Cockerill.....	1	4	s.	sc.	83,000	25,000	58,500	540	706	1.81
250	150	Cockerill.....	2	4	s.	td.	65,000	10,000	23,400	300	353	1.24
300	120	Deutz.....	1	4	s.	sc.	83,500	35,000	81,800	295	551	2.07
300	120	Deutz.....	2	4	s.	tw.	161,000	14,000	32,800	383	447	1.52
300	140	Deutz.	4	4	s.	d. tw.	110,000	3,500	8,200	484	394	1.32
600	80	Cockerill.....	1	4	s.	sc.	207,000	100,000	234,000	512	734	0.99
600	130	Cockerill.....	2	4	s.	td.	185,000	46,000	107,500	386	487	1.13
600	110	Oechelhaeuser	1	2	s.	sc.	143,000	48,000	112,000	318	425	1.23
600	130	Deutz.....	2	4	s.	tw.	158,000	28,000	65,500	310	371	1.67
600	130	Deutz..... ...	4	4	s.	d. tw.	189,000	7,000	16,400	327	342	1.08
600	110	Koerting.....	1	2	d.	sc.	136,500	18,000	42,200	258	297	1.11
750	90	Nuernberg.....	1	4	s.	sc.	297,000	115,000	26,900	560	538	1.03
1200	80	Cockerill.....	2	4	s.	td.	365,000	95,000	222,000	383	488	0.68
1200	130	Deutz.....	4	4	s.	d. tw.	354,000	14,000	32,800	307	322	1.01
1200	120	Nuernberg.....	4	4	s.	d. tw.	280,000	16,000	37,400	246	264	0.94
1200	110	Oechelhaeuser	2	2	s.	tw.	260,000	16,000	37,400	230	248	0.9
1200	110	Koerting	2	2	d.	tw.	250,000	4,500	10,500	212	218	0.9
1400	110	Cockerill.. ...	2	4	d.	td.	374,000	8,600	20,000	170	164	0.42

A comparison of columns 11 and 12 is particularly interesting, as it will prove that for lower regularity and for small units, the engine with the fewer cylinders builds lighter than that with more cylinders; but, on the other hand, that the former cannot come into question for high regularity and for larger power units. In these columns the comparison is based upon engines driving blowing cylinders and engines driving direct-current generators running in parallel. The fly-wheel differences. An idea of the advances which have been gradually made is best gathered from Table 2, in which the gases of various compositions are compared on a basis of their heat value. In this list

Column 1 is gas consumption in cubic feet per brake horse-power-hour.

Column 2 is heat value of gas in B.t.u. per cubic foot.

Column 3 is B.t.u. per brake horse power-hour.

TABLE 2. SHOWING GAS CONSUMPTION, HEAT VALUE IN B.T.U., AND B.T.U. PER B.H.P.-HOUR

	1	2	3
1. Deutz engine, in Freidenshutte	135	100	13,500
2. Cockerill engine, in Seraing	116.5	110	12,800
3. Oechelhaeuser engine in Hoerde	107.5	103	11,050
4. Koerting engine, in Hannover	87.6	130	10,620
5. Anhalt engine, in Differdingen	95.2	101	9,620
6. Premier engine, in Winnington	70	135	9,100

These figures indicate that for German conditions it is perfectly safe to figure upon a gas consumption of 106 cubic feet of blast-furnace gas per brake horse-power-hour, and from conservative comparison, it may be assumed that under American conditions about 120 to 125 cubic feet would be required. Those figures are all based upon engines running under normal load. They become very much less favorable when the engine is under-loaded, and the gas consumption per brake horse-power-hour varies under different loads about as follows, the figures being calculated for American conditions:

With overload of 25 per cent., gas consumption, 115 cubic feet per B. H. P.-hour.
With full load, gas consumption, 122 cubic feet per B. H. P.-hour.
With 80 per cent. load, gas consumption, 137.5 cubic feet per B. H. P.-hour.
With 50 per cent. load, gas consumption, 163 cubic feet per B. H. P.-hour.
With 25 per cent. load, gas consumption, 200 cubic feet per B. H. P.-hour.

Should the load decrease to zero, so that the engine runs idle, the gas consumption would increase still further, because the friction of the gas engine is high. The following figures may be considered as averages:

A gas engine (with proper charge distribution) designed to develop normally 600 brake horse-power will show about the following consumption of blast-furnace gas:

When developing 750 B. H. P., 86,200 cubic feet per hour.
When developing 600 B. H. P., 73,300 cubic feet per hour.
When developing 450 B. H. P., 61,800 cubic feet per hour.
When developing 300 B. H. P., 49,000 cubic feet per hour.
When developing 150 B. H. P., 25,000 cubic feet per hour.
Friction load only, 25,000 cubic feet per hour.

The main consideration affecting the consumption of lubricating oil and cooling water are the dimensions of the engine. Engines with large cylinder diameters are, in this respect, more wasteful than the smaller units, and consequently the Cockerill engines are at a disadvantage. The majority of these require about 20 gallons of cooling water per brake horse-power per hour; in other words, a 600-brake-horse-power engine running continuously under normal load will require about 290,000 gallons of water per twenty-four hours. The consumption of lubricating oil varies between 0.0045 and 0.0055 pint per brake horse-power-hours; in other words, the 600-brake-horse-power engine running under normal load will consume from 8 to 10 gallons per twenty-four hours. The quantities will be less for engines of smaller cylinder diameters, but of the same power, for instance, in two-cycle and double-acting machines the consumption of cooling water is about 10 gallons, and of lubricating oil about 0.0035 pint per brake horse-power-hour, on which basis a 600-brake-horse-power engine under normal load will require 150,000 gallons of cooling water and 6 gallons of lubricating oil per twenty-four hours.

The consumption of water and oil does not depend entirely upon the careful construction of the engine, but to a fully equal extent upon the ability and the good will of the engineer. In any event, the consumption will be considerably higher in new machines, until same have found their bearings; especially that of the lubricating oil, which will, at the start, be twice or three times as much as above stated.

Efficiency

In connection with this subject, it is desirable to give brief consideration to the mechanical efficiency of the gas engines. This depends upon the care of the attendants fully as much as upon the design and construction. The following figures may, however, be taken as averages:

Single-acting four-cycle engines:
One-cylinder, 85 to 90 per cent. efficiency.
Two-cylinder, 80 to 85 per cent. efficiency.
Four-cylinder, 75 to 85 per cent. efficiency.

Data covering the double-acting four-cycle engines are not available.

Single-acting two-cycle engines:
One cylinder, 78 to 82 per cent. efficiency.

Double-acting two-cycle engines:
One cylinder, 70 to 75 per cent. efficiency.

On this basis a 600-brake-horse-power engine will consume in internal friction the following amounts, depending upon its construction:

Single-acting four-cycle with one cylinder about 90 horse-power.

Single-acting four cycle with two cylinders, about 130 horse-power.

Single-acting four-cycle with four cylinders, about 150 horse-power.

Single-acting two-cycle with one cylinder, about 150 horse-power.

Double-acting two-cycle with one cylinder, about 225 horse-power.

These figures are much higher than in good steam engines, and as the engine friction under no load is only a few per cent. lower than under full load, it exercises a very considerable effect upon the gas consumption, and this explains the rapid increase of the latter at under-loads indicated in the preceding figures. Economy in gas consumption under varying loads is, therefore, not obtainable merely by the complete utilization of the fuel, but to attain this end, there is still room for mechanical improvement.

As above stated, the mechanical efficiency of the gas engine is very much lower than that of a steam engine; nevertheless the total efficiency from a thermodynamic standpoint is very much higher in a gas-engine plant than in a steam-engine plant, and in this connection it will be instructive to compare the gas consumption of a steam-engine plant developing the same power and using waste gases for fuel under the boilers.

For the steam engine, a steam consumption of the entire plant—that is, engine and condensing outfit—is figured at 17 pounds per engine brake horse-power-hour, and the total efficiency of the engine and condenser pumps 87 per cent. at normal load; the steam losses due to boiler radiation, pipe friction, and condensation 8 per cent. of the total normal steam consumption, the boilers generating one pound of steam per 25 cubic feet of gas consumed. No special consideration is given in these figures to gas losses in the mains, as this would be about the same between the cleaning plant and the gas engines, as between the cleaning plant and boilers. The total gas consumption in cubic feet per hour will be as follows:

B. H. P.	Gas Engine.	Steam Engine.
750	86,200	330,400
600	73,300	262,400
450	61,800	188,400
300	49,000	159,400
150	30,500	98,400
000	25,000	42,100

While these figures are for waste blast-furnace gas, it may be stated that a fair value of the heat consumption of an engine operating on any gas will be between 10,000 and 13,000 B.t.u. per brake horse-power per hour, depending on the size of the engine. In some of the larger engines the figure has been reduced to 9000 B.t.u. and even less.

The University of Illinois at Urbana, Ill., recently established ten "research fellowships" in its engineering experiment station, which was established in connection with the College of Engineering in 1903. These fellowships have an annual value of $500, and are open to graduates of approved universities and technical schools, both American and foreign. Preference will be given to men who have had some experience in practical engineering work outside of college. The object is to develop experts along various lines of engineering, some of whom may be attached later to the regular corps of the station.

The speed of the crank-pin is 1.57 times that of the piston. If the piston speed is given, as usually in feet per *minute*, and it is desired to find the crank-pin velocity in feet per *second*, multiply the piston speed by 0.026. The velocity of the crank-pin in feet per second may be found by multiplying 0.0136136 by the stroke in inches and the revolutions per minute.

The Hydraulic Elevator—V

Further Discussion of the Simple Low-pressure Type, Explaining Lever Control, Regulating Valves and Safety Devices

BY WILLIAM BAXTER, JR.

In the March number was discussed the vertical-cylinder elevator of the type used in buildings from five to seven or eight stories high. The same style of elevator could be used for higher runs, by increasing the gear to three or four to one, but it is not desirable to do so, owing to the fact that as the hight of the building increases the car speed is increased, and hand-rope control is not satisfactory when the car runs faster than about 250 feet per minute. When a car runs at a low velocity the operator can control it perfectly by means of the hand rope; but as the velocity increases, the difficulty of making accurate stops at the floors of the building becomes so great that only the most experienced elevator operators can control the movement of the car in a manner to give satisfaction to the passengers. Furthermore, the difficulty of controlling a high-speed car by means of a hand rope is due not only to the increased velocity but also to the fact that more effort is required to move the valve, as the latter must be increased in size, owing to the fact that more power is required to develop the higher speed. The effort to move the hand rope can be kept down somewhat by increasing the distance through which it must be moved to open or close the valve, but this does not help matters very much.

Even if the difficulty of car control could be entirely overcome, the hand rope would still be objectionable for high speeds, because it would be necessary to adjust the automatic top and bottom stops so as to retard the car too rapidly to make a satisfactory stop or else give the hand rope so much movement that the operator could not make at intermediate floors as quick stops as are often required. For these reasons the hand-rope control is not used in high-speed elevators intended for first-class service.

In place of the hand rope an operating lever is used, and as the movement of the end of this is only through a distance of about one foot, it would be next to impossible for the operator to move the operating valve "by," for it would require an effort that even an athlete could not exert for more than a few minutes. To get around this difficulty, a small valve operated by the car lever is provided, and this valve controls the flow of water in and out of a motor cylinder, the piston of which moves the main operating valve. This small valve is called a "pilot," and elevators in which it is used are said to be provided with "a pilot-valve and car-lever control." An elevator of this type, as made by the Otis Elevator Company, geared four to one, is shown in Fig. 34.

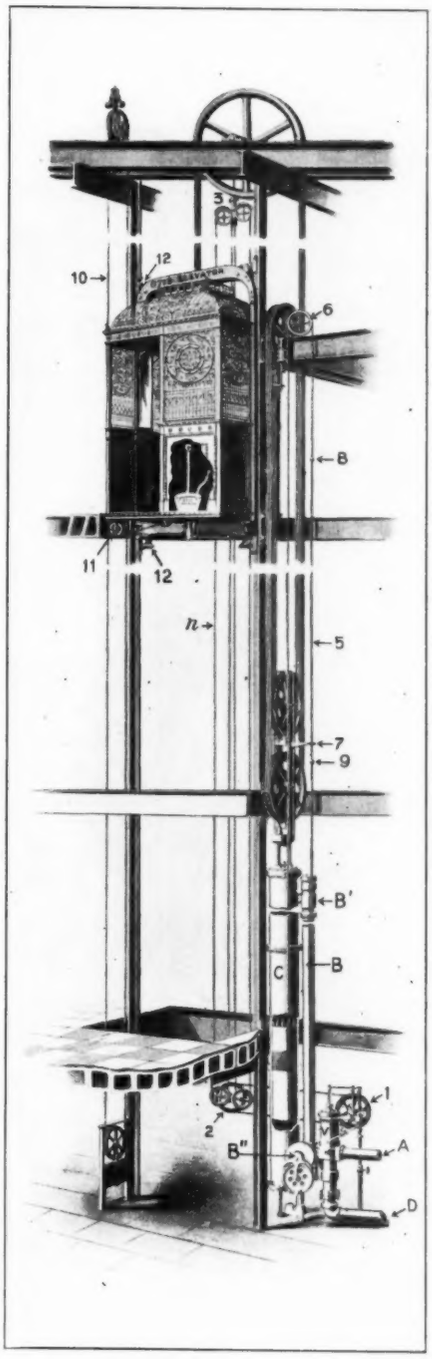


FIG 34. RUNNING-ROPE TYPE, SHOWING PILOT-VALVE AND CAR-LEVER CONTROL

It will be noticed in Fig. 34 that there is an extension at the top of the valve chamber, on the right side. This extension carries the pilot valve and also a sheave *1*. This sheave moves the pilot valve and the movement of the latter controls the flow of water into and out of the upper end of the main valve chamber, which, as will be seen, is of larger diameter than the lower portion. In this enlarged end of the valve chamber is located a piston that moves the main valve. The sheave *1* is rotated through a small angle, in either direction, by sheave *2;* the latter carries on one side two smaller sheaves over which pass ropes *n* that run up the elevator well and at the top of the building pass over two similar sheaves which are stationary. These ropes are connected with a lever mounted on the rear end of a shaft running under the elevator car, and which carries at its front end a lever that projects up through the floor of the car. This is the operating lever, and by turning it in one direction the sheave *1* is caused to rotate in a corresponding direction, say in the direction that will move the pilot valve so as to cause the car to ascend. Moving the car lever in the opposite direction will cause the car to descend.

The pressure water enters through pipe *A* and is discharged through pipe *D*, in the lower end of the cylinder. At *B'* is placed a speed-regulating valve, the office of which is to prevent the car from running at too high a velocity if the operator throws the main valve wide open when the load is light. At *B"*, between the main valve and the lower end of the cylinder, is located a valve which stops the car automatically at the top and bottom floors of the building. This valve is actuated by the rope *5*, which passes through the end of arm *7* projecting from one side of the traveling-sheave frame. The rope passes over a sheave *6* and thence runs down and around a sheave mounted on the back end of a shaft carrying a pinion which meshes into the gear on the automatic stop valve *B"*.

The illustration shows a car running on iron guides, which are the only kind permitted in fireproof buildings. For metal guides the wedge safety shown in Fig. 25 in the March number cannot be used, as the friction of wood against iron is too uncertain, being considerable if there is any grit between the surfaces, and slight if there is no grit and the surfaces are well lubricated. The safety shown in Fig. 34 is of the type known as a clamp, or brake safety. It is also called a drum safety, or a toggle-joint safety. It is

given all these names because it is a brake made in the form of heavy shoes that are clamped tightly against the elevator guides by means of the toggle joint, or equivalent devices, operated by the rotation of a drum placed in the safety at a point between the two ends.

The construction and operation of the several parts of Fig. 34 can be better explained in connection with the line drawing Fig. 35, and in the other line drawings which follow. In Fig. 35 the speed-regulating valve B' is not shown, and the rope connections for transmitting the motion of the car lever to the pilot-valve sheave *1* are not the same. There are many modifications of these rope connections, but all of them are classified under two generic systems known as the "running rope" and "standing rope." The arrangement of Fig. 34 is of the running-rope type, while that of Fig. 35 is of the standing-rope.

The Standing-rope System

The operation of the standing-rope system can be more easily understood from Fig. 36, which shows the rope connections complete and free from other complications. The rope *n'* comes down from top sheave *3* on the right, and passes under the right-side sheave carried by the cross lever connected with the car-operating lever. From the under side of the right-side sheave the rope passes to and around the top of the left sheave, and thence down and under the two lower stationary sheaves *2, 2*, then up and around the pilot-valve operating sheave, and under the two central stationary sheaves *2, 2*, thence over the right and under the left sheaves carried by the car lever, and so on up over top sheave *3*, where the two ends of the rope are connected and from their end is suspended the tension weight *3'*. If the car lever is moved to the right, the ropes *n* will slacken up while the ropes *n'* will draw up, from the fact that the right-side sheave will be depressed and the left-side one elevated. This change in the ropes will rotate the pilot-valve sheave clockwise. If the car lever is moved to the left, the action will be just the opposite and the valve-operating sheave will be rotated counter-clockwise. Thus, through the taking up of one rope and the letting out of the other, the pilot-valve operating sheave is made to move in accord with the movement of the car lever. This system, which as already explained is made in several modified forms, is called the standing rope, because the ropes remain stationary while the car is running. In running-rope systems, the ends of the ropes are fastened to the car lever, or two ends to the lever and two to the car, and the ropes run when the car runs.

Returning to Fig. 35, it will be seen that when the sheave *1* is rotated clockwise, the lever *4* is pulled down by the connecting-rod *13*, the lower end of which is connected with the crank on the sheave shaft. This downward movement of *4* will depress rod *14*, and thus move downward the pilot valve V^2, simply because it is easier to move this end down than the opposite end which is connected with the top of the main valve-rod. When the pilot valve is moved down, pressure water will flow from pipe *A* through the small pipe *A'* to and through the valve to pipe *A'*, and thus to the upper end of the main valve chamber, above the motor piston *V'*.

The pressure water is at all times in the space between valve *V* and the under side of *V'* and also in the space above *V'*, but so long as water cannot pass in or out of this latter space the main valve cannot move, for it could not move upward without compressing this water, and it could not move downward without forming a vacuum above *V'*. As there is pressure water above and below *V'*, this valve taken by itself is in a state of perfect balance, but the main valve *V* is unbalanced, because the pressure water acts against its upper side, while only the pressure of the discharge tank acts against its lower side. As a

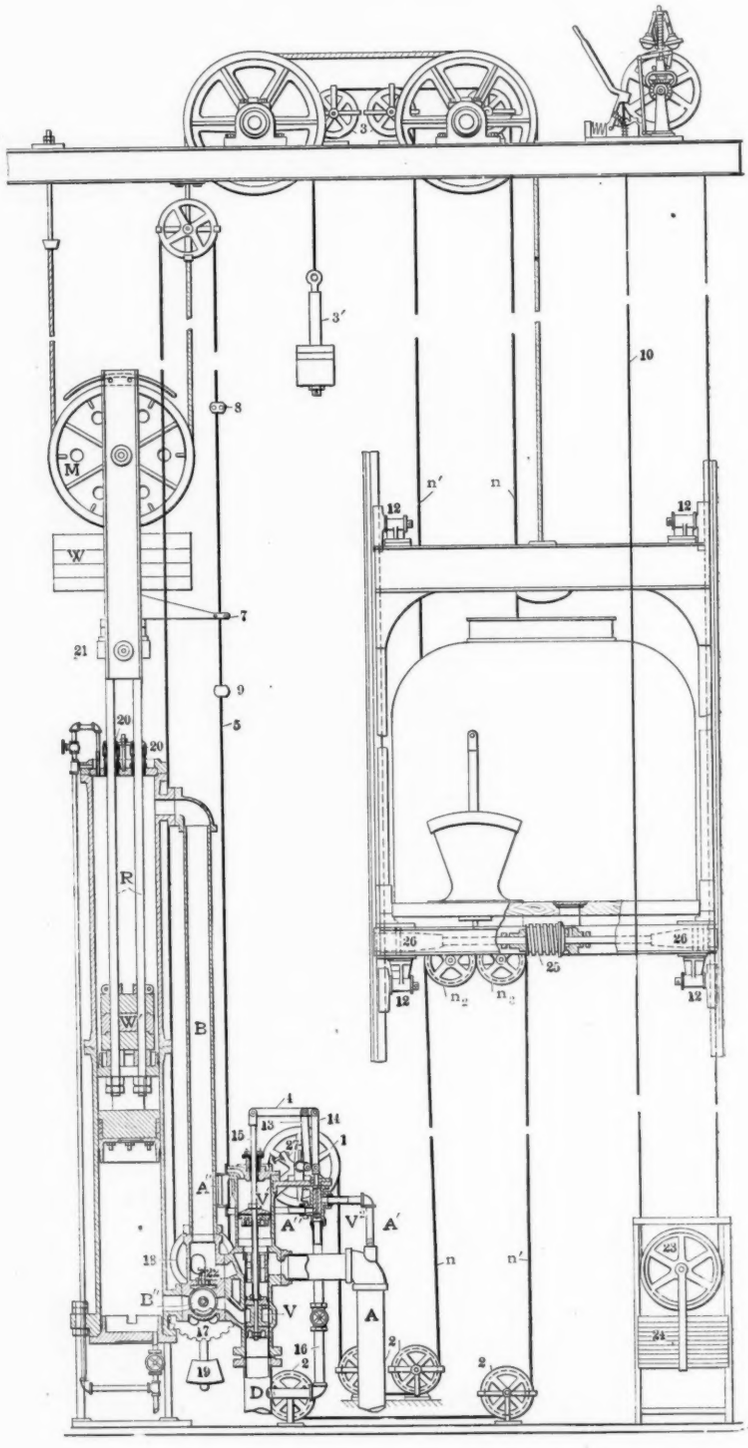


FIG. 35. PILOT-VALVE AND CAR-LEVER CONTROL, STANDING-ROPE TYPE

result of this state of things, as soon as water enters the space above V', this piston will move down, carrying the main valve with it, thus connecting the top and bottom of the cylinder, so that the water may circulate and the piston run upward. It will be noticed that as soon as V' begins to move down by the admission of water above it, the pilot valve is moved upward, because rod *13* will be held stationary by sheave *1*. When V' moves

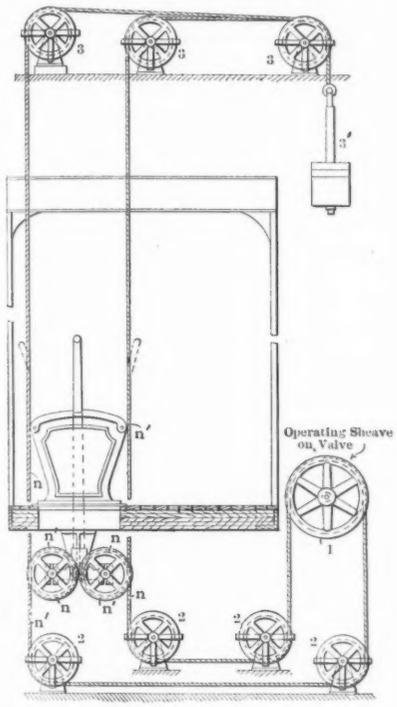


FIG. 36. ILLUSTRATING OPERATION OF STANDING-ROPE SYSTEM

down a certain distance, it will have moved the pilot valve upward to the central or stop position; that is, the position in which the flow of water into the space above V' is stopped; hence, when this point is reached, V' will descend no farther.

If the sheave *1* is turned counter-clockwise, rod *14* will be lifted and thus pull up the pilot valve V^2, and this movement will connect pipe A'' with pipe *16* and permit the water in the space above V' to run out into the discharge pipe D, and the main valve to ascend and connect the lower end of the cylinder with the discharge pipe; so that the piston may run down and pull up the elevator car. In this movement it will again be seen that as soon as V' begins to move upward, it will move rod *14* downward, and thus depress the pilot valve until it reaches the stop position and closes off the connection between A'' and the pipe *16*. To fully understand this movement of V' it must be remembered that the pressure in pipe *16* is much below that in the pipe A, so that when A'' is connected with *16* the pressure above V' is greatly reduced and the latter being unbalanced will be forced upward by the pressure under it, which will be greater than the downward pressure on V, owing to the difference in the diameters of the two pistons.

The distance through which the sheave *1* is rotated depends upon the distance through which the car-operating lever is moved, so that if the latter is moved a short distance the sheave *1* will rotate through a small angle. In this case the pilot valve will be shifted but a short distance from the central position, and, therefore, the main valve will only have to move a small portion of its full stroke to return the pilot to the central position. If the car lever is moved as far as it will go, the rotation of the sheave *1* will be through a wide angle and the pilot will be shifted a considerable distance from the central position. In this case the main valve will have to move its full stroke to return the pilot to the central position. Thus it will be seen that in every case the opening of the main valve is directly proportional to the distance through which the car lever is moved, and the control the operator in the car has over the movement of the main valve is as complete as it would be if he stood by the side of the valve and effected its movement by means of a lever attached directly to the piston-rod *15*.

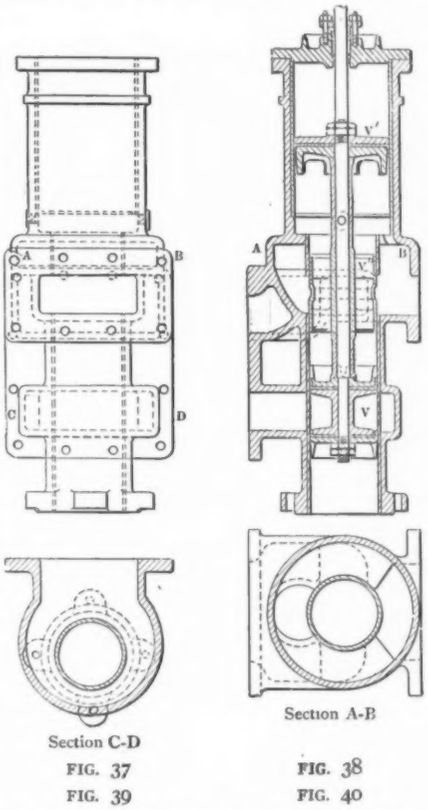
Pilot-valve Control

All modern hydraulic elevators intended for first-class passenger service are provided with a pilot-valve control, and although the design in the various types may differ considerably, the principle of operation is the same as that of the valve here shown. This design, however, is the best one from which to gain a thorough knowledge of the principles of operation, because it shows them up very clearly. The stop screw *27* is a safety feature provided to prevent the operator from opening the valve wide enough to permit the car to run away when running up with a light load. When the speed regulator B', shown in Fig. 34, is used, this stop simply becomes an additional precaution, as B' alone is capable of taking care of the car speed, if properly adjusted.

The construction of the main valve of Fig. 35 can be easily understood from the drawings Figs. 37 to 40. Fig. 37 is an external elevation taken from the right side of Fig. 38, the latter being a vertical elevation in section, showing the valve in the stop position. Fig. 39 is a horizontal section on the line C—D through the port that connects with the lower end of the cylinder, and Fig. 40 is a horizontal section taken on line A—B, just above the port with which the pressure pipe A is connected. The lower portion of the valve is the same as that shown in the March number, and so is the corresponding part of the valve chamber.

The upper part differs from the last-named valve simply in that it is of a larger diameter. In the simple hand-rope valve the pistons V and V' are made of the same diameter, so that the pressure of the water pushing up against V' may be just equal to the pressure pushing down upon V, to produce a perfect balance and make the valve as easily movable upward as downward, disregarding the weight of the valve itself. In the valve here shown it is necessary to make V' of a larger diameter than V, so that when the pressure above V' is reduced the valve may become unbalanced and thus be forced upward by the pressure acting on the under side. This valve is provided with the throttle valve V'', which will prevent a runaway if for any reason the pressure in the tank drops to zero, or nearly so, and, as fully explained in March POWER, also prevents making a violent stop on the up trip if the operator moves the lever to the central or stop position too suddenly.

The construction of the pilot valve is well shown in Figs. 41 and 42, the first being a sectional elevation of the valve and casing, showing the valve in the stop position, and the second a sectional elevation of the casing alone, taken at right angles to Fig. 42. The valve consists of a sleeve mounted upon the spindle V_s, which serves to hold in place cup packings *40*, *41*



Section C-D Section A-B

FIG. 37 FIG. 38

FIG. 39 FIG. 40

EXTERNAL AND SECTIONAL ELEVATIONS OF MAIN VALVE

and *42*, as clearly shown. The pressure water enters through the port *45*, and passes to the pipe A'' through the port *46*. The discharge water from the upper end of the main valve chamber enters through the port *46* and discharges through the lower end *47* into the pipe *16* (see Fig. 35). The valve chamber is provided with brass linings,

and the lower lining has numerous holes drilled through it opposite the port *46*, so that the cup packings may pass over them on the up stroke without being caught.

In Fig. 41 it can be seen that the packings only lap a short distance over the port holes drilled in the brass lining; therefore, the valve does not have to be moved very far in either direction to uncover the ports and start the elevator. The amount of lap of the valve packings is made sufficient to cause the valve to close when the car lever is some distance from the central position. If the valve were made with less lap, the operator would have to move the lever with great accuracy to stop the car, for if it were not moved far enough the car would slow down to a very low speed, but would not stop; and if the lever were carried a trifle further the car would stop and then begin to run in the opposite direction. To obviate this trouble the valve packings are made to lap enough to require a considerable movement of the car lever to either side of the center to open the ports, but not any more than is actu-

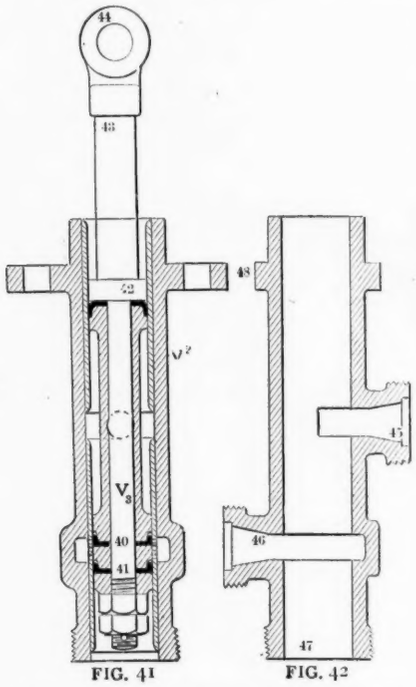


FIG. 41 FIG. 42

SHOWING CONSTRUCTION OF PILOT VALVE

ally necessary for the proper operation of the car.

In the simple hand-rope control elevator the car is stopped automatically at the top and bottom landings by the closing of the main valve through the movement of the hand rope by the motion of the car itself, the latter striking stop balls fastened to the rope at the proper points. In the pilot-valve control, as can be seen in Fig. 35, a separate stop valve *B″* is placed between the valve chamber and the lower end of the cylinder. This valve is actuated by the rope *5* which has stop balls fastened to it at proper points, as shown at *8* and *9*. These balls are struck by the arm *7* when the car comes within proper distance from either end of its travel for the automatic valve to start to close. The closing of this valve is gradual so that it does not entirely check the flow of water until the car has traveled a distance that may vary from about three feet up to six, eight or more feet, depending upon the speed at which the elevator runs. The distance within which the valve is closed is determined by the size of the sheave *18*, the gear *17*, and a pinion that drives *17*.

Looking at Fig. 35 it can be seen that if the car is going up the arm *7* will descend and at the proper time will strike the stop ball *9*, and thus carry the rope *5* with it so as to rotate the sheave *18* in a clockwise direction, and the pinion of the sheave-shaft will rotate *17* and therefore the valve *B″* in a counter-clockwise direction, so that the valve will stop off the flow of water from under the piston into the pipe *D*, by swinging up and over the port on the right side of the circular valve chamber. If the car is running up at full speed, and is stopped entirely by the action of *B″*, it may shut off the flow of water faster than the momentum of the moving parts is overcome by the action of gravity, so that the car may continue moving after the valve is closed. In such a case the car ropes would tend to slack up as the piston could not move any further, owing to the incompressibility of the water. In a case of this kind there would be no such occurrence with the valve *B″* as constructed, because the relief valve *22* would be raised, and the water imprisoned under the piston would find an outlet into the lower end of the circulating pipe, and thus to the upper end of the cylinder.

It is very seldom that the valve *22* is called into action through the too sudden stopping of the car by the action of the valve *B″*, as this valve is always adjusted so as to not act so quickly as to produce such an effect, but sometimes valves can get out of adjustment. The principal value of the relief valve *22* is to prevent too sudden stops when the operator pulls the lever to the stop position too rapidly. This valve is used even with the simple hand-rope control, as was mentioned in the March number, the casting which forms the foot of the circulating pipe being of the form shown in Fig. 43, which differs from the casting shown in Fig. 35 only in not having a circular chamber to accommodate the valve *B″*.

Automatic Stop Valve

The construction of the automatic stop valve is fully shown in Figs. 44 and 45, the first of which is a view taken at right angles to the axis of the shaft, with the right side in section. The other drawing is a view parallel with the valve-shaft, and also shows the right-side half in section. The face of the valve is formed by the segment *50*, which is held in position and moved by three-flanged hubs, *49*, these latter being firmly secured to the shaft *53*. The gear *17* is mounted on this shaft, and is driven by the pinion *55*, which is cast on one side of the sheave *18*. The segment *50* rotates within a brass lining *51*, which has ports cut through it on both sides, the upper edge of the ports being on an inclined line, as shown at *52* in Fig. 45; this construction being for the purpose of closing the port gradually so as to not stop the elevator too suddenly. The segment *50* is not rigidly secured to the hubs *49* for the reason that, if so made, it would not be possible to start the elevator from either end of the well.

This valve moves into the position to close the outlet from the lower end of the cylinder, at the end of every trip, and is held in that position by the arm *7* bearing against one or the other of the stop balls, *8*, *9*; therefore, unless the piston moves away from the end far enough to permit the stop ball to move the valve *B″* will remain closed, so that the opening of the main valve would not have any effect in starting the eleator. With the segment *50* made so as to have considerable play

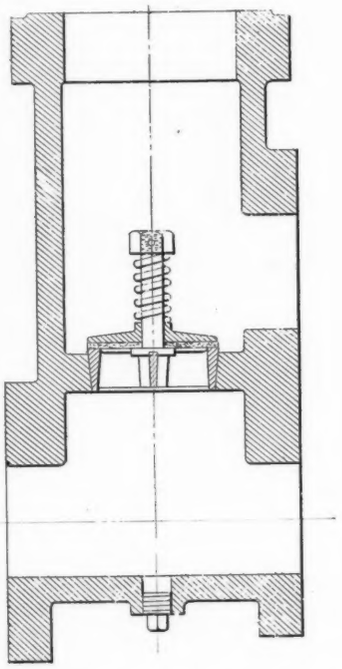


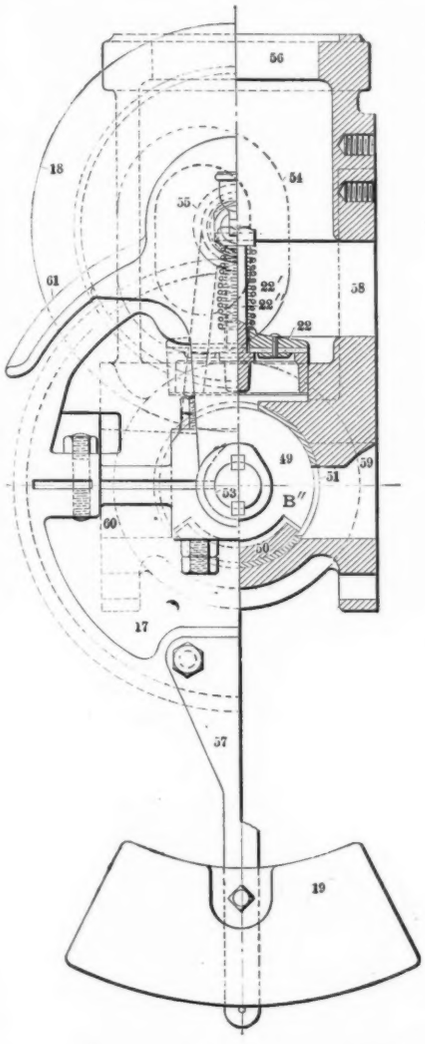
FIG. 43. SECTIONAL ELEVATION OF PILOT-VALVE CASING

in the supporting flanges of the hubs *49*, as is clearly shown in Fig 44, there is no difficulty in making a start, because when the elevator is stopped, the current of water flowing through *B″* is in such a direction as to press the segment *50* against the casing *51*, and when the main valve is moved so as to run the elevator in the opposite direction, the pressure of the water acting on *B″* will be toward the center of the circle and thus will push *50* away from *51*, and permit a sufficient amount of water to leak through to start the car gradually. As soon as the car begins to move, the arm *7* permits the stop ball to return to the normal position and

then the weight *19* draws the gear *17*, and thereby the valve *B″* into the open position. From this it will be seen that this valve not only acts as an automatic limit stop, but also it provides means whereby the car is prevented from starting off with a violent jerk if the operator moves the lever too rapidly to the full-speed position.

Speed-regulating Valve

The construction and principle of action of the speed-regulating valve *B′*, shown in Fig. 34, will be more fully understood by reference to the line drawing Fig. 46, which is a section parallel with the axis of the valve-rod. The port *63* is bolted against the upper end of the cylinder, while the circulating pipe is screwed into the lower end *62*, thus whether water is passing from the pressure tank into the upper end of the cylinder, or the water is simply circulating from the upper end to the lower end of the cylinder, it must pass through this speed-regulating valve. Suppose the elevator is running upward, then water will pass from the pressure tank, up through the regulating valve to the upper end of the cylinder. This current of water will strike against the lower end of the valve *70*, and push it upward against the tension of the spring *71*. If the elevator is running down, the stream of water will flow from the port *63* down through the regulator and into the circulating pipe. In this case the force of the water will act against the top of the valve *70* against the tension of the spring *72*. By properly adjusting these springs the device can be arranged so as to act at any desired velocity. The action depends entirely upon the velocity of the water flowing through the valve; therefore, it is not affected by any variations that may occur in the pressure of water. The operation of the valve is so self-evident as to require no very extended explanation. It can be seen that on the down stroke of the piston, if the valve is forced upward far enough by the impact of the stream of water, the upper end will enter the neck *73*, and thus gradually reduce the opening through which the water must pass, and thereby reduce the volume that passes, hence the velocity. On the up-stroke of the piston, the valve is forced down into the neck *74*, of the piston in the cylinder in the same manner. The interior valve *67* is for the purpose of preventing too sudden a movement of the valve *70*, that is, it acts as a dash-pot. The packings *68* make a tight joint, and the proper amount of leakage to give the desired retardation is obtained by providing the necessary difference between the diameter of the rod *64* and the holes through the ends of the valve. The springs *71* and *72* can be compressed more or less, as may be required, by means of the adjusting sleeves at their ends.



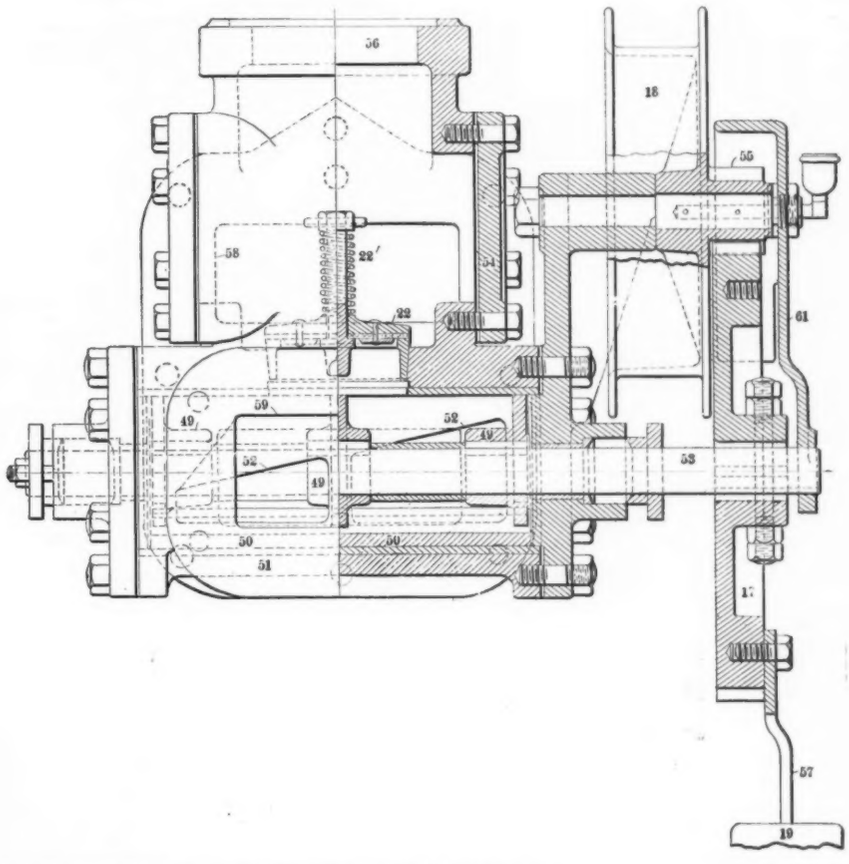


FIG. 44 CONSTRUCTION OF AUTOMATIC STOP VALVE FOR HYDRAULIC ELEVATOR FIG. 45

Safety Devices

The safety device shown on the car in Figs. 34 and 35 is of the clamp, or brake type. It is known as the "Otis wedge-clamp safety," and its construction and operation can be fully understood from Fig. 47 and the line drawings that follow. The lower part of Fig. 47 shows the under side of the elevator car, with the safety attached thereto. The center portion shows the roof of the car and the device placed thereon to hold the governor rope. The upper portion shows the safety governor, whose office is to actuate the safety. For the purpose of showing clearly the construction of the safety, the side of the framing that is in front of these parts is drawn as if it were transparent, and while this type of illustration serves admirably to accomplish the object intended, it is liable to be misleading if it is not kept in mind that this side is made of a channel beam so that in the actual apparatus the internal parts cannot be seen at all. The operation of the safety is as follows:

The end pieces *26* form powerful clamps that are pressed against the same elevator guides that the guide-shoes *12* run on. At the inner ends of these clamps rollers *80* are mounted and between these wedge-shaped pieces *79* are forced by the

rotation of the drum *25*. The forcing of these wedges between the rollers *80* spreads the inner ends of the clamps *26* and forces the outer ends against the elevator guides; thus developing a frictional resistance that increases gradually as the pressure of the clamps is increased, until the car is brought to a stop. The rotation of the drum *25* causes the shafts *78* to move outward, and thus force the wedges *79* into the space between the rollers *80*. The drum *25* is rotated by the rope *11*, and this rope is actuated by the safety governor in the following manner: The governor rope *10* is in effect endless, as shown in Figs. 34 and 35, its ends being fastened to the piece *10'*. This piece has conical projections on its sides that fit into depressions in the ends of the levers *10"*. These levers are pivoted at 10_1, and their back ends are forced apart by a spring 10_3. The tension of this spring is sufficient to hold the levers *10"* in engagement with *10'* firmly enough to drive the safety governor; so that in the normal running of the elevator the parts remain in the position in which they are shown, and the governor is driven at a speed that corresponds with the velocity of the car.

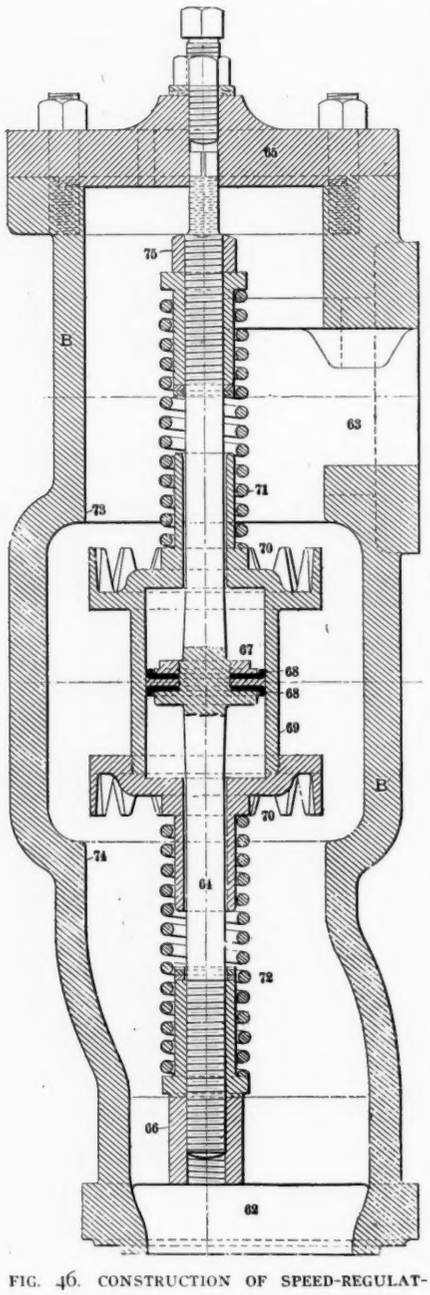


FIG. 46. CONSTRUCTION OF SPEED-REGULATING VALVE

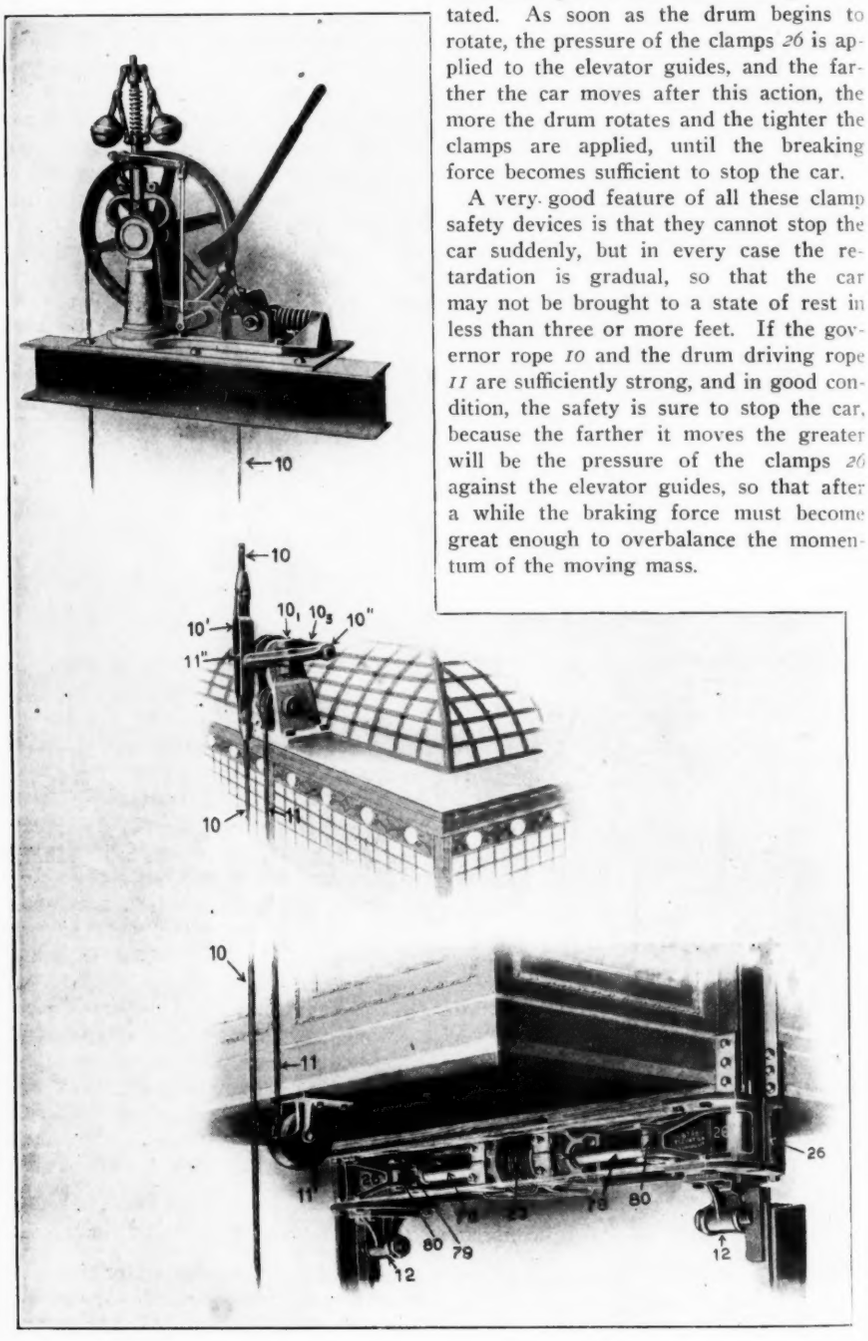


FIG. 47. ILLUSTRATING THE OTIS WEDGE-CLAMP SAFETY DEVICE

The stand 10_2 holds two small sheaves, in addition to the levers *10"*, and the rope *11* passes over the lower sheave and around the top one, its end coming forward and being firmly secured to *10'*, at the point *11"*, as shown in the illustration. If the elevator car attains a velocity higher than that for which the governor has been adjusted, the balls will swing outward so far that the lifting-rod will throw the rope clamps into action, and thus immediately stop the movement of the latter. As the car still continues to move, the piece *10'* is pulled away from the ends of levers *10"* and then the rope *11* is drawn upward and the drum *25* is rotated. As soon as the drum begins to rotate, the pressure of the clamps *26* is applied to the elevator guides, and the farther the car moves after this action, the more the drum rotates and the tighter the clamps are applied, until the breaking force becomes sufficient to stop the car.

A very good feature of all these clamp safety devices is that they cannot stop the car suddenly, but in every case the retardation is gradual, so that the car may not be brought to a state of rest in less than three or more feet. If the governor rope *10* and the drum driving rope *11* are sufficiently strong, and in good condition, the safety is sure to stop the car, because the farther it moves the greater will be the pressure of the clamps *26* against the elevator guides, so that after a while the braking force must become great enough to overbalance the momentum of the moving mass.

The construction of the several parts of this safety can be better understood from the line drawings Figs. *48* and *49*, the first being a side view and the second a plan. The brake clamps *26* are hinged on a strong steel stud *81*, and carry at their inner ends rollers *80*. The wedge-shaped pieces *79* are mounted on the ends of the shafts *78* and are prevented from getting out of line by side guides. The shafts *78*

are prevented from turning by pins that slide in slots in the bearings in which they are held, as is shown in Fig. 49, and also in Fig. 47. The inner sleeves of the drum are threaded to fit the screw ends of the shafts *78*, with right- and left-hand threads, so that when the drum rotates the shafts are forced outward, and the wedges *79* are driven in between the rollers *80*, thus forcing the outer ends of the clamps *26* together, with an ever increasing force. The bevel gear *82* at the end of the drum is for the purpose of releasing the clamps after the safety has gone into action. This gear is used in connection with a wrench that consists of a pinion *83*, shown in broken lines in Fig. 48, mounted on the lower end of a stem *84* that is constructed at the top with a handle *85*. A trap-door is provided in the car, and by removing this the wrench can be put in position and by means of it the drum can be turned backward to the normal position so as to release the clamps *26*.

Fig. 48 also shows the construction of the elevator guide-shoes *12*. As will be noticed, these are provided with adjusting screws, and as they work on a swivel, they can be set so as to have but little play. On this account the brake clamps *26* can also be set close without fear of their rubbing against the elevator guides and thus getting so loose that the clamp will not act soon enough, or with sufficient force when called into action.

In Fig. 34 it will be noticed that there is a second lever in the car on the right side of the operating lever. This is an emergency lever used in most first-class elevators to enable the operator to retard or even stop the car, if for any reason he cannot control it by means of the regular lever. The construction of this device (which is simply an addition to the clamp safety) and its operation can be understood from the drawings Figs. 50, 51 and 52, the first of which is an elevation giving a side view of the main and the emergency levers. Fig. 51 is an elevation at right angles to Fig. 50, looking at the latter from the left side, and gives an edge view of the two levers. Fig. 52 is a plan of one-half of a car, showing the safety device and the connecting-rods and links connecting it with the emergency lever, in broken lines. In this arrangement, the shafts *78* are made in two parts and are joined by the hubs of the cranks *94*, the latter having right- and left-hand threads in their bore, into which the ends of *78* are screwed. When the emergency lever *86* is moved toward the position *86'*, it acts through the connecting levers and rods to turn the cranks *94* in the direction that will force the ends of *78* out, spread the wedges *79* and apply the brake clamps *26*. The several cranks and connecting-rods are numbered consecutively, so that following the numbers the way in which the motion of *86* is transmitted to *94* can be easily traced. This device can be used to stop the car if the lever *86* is moved far enough, but it would be better to apply it with enough force to simply retard the velocity of the car to, say, half speed or less, and thus descend safely to the bottom floor of the building.

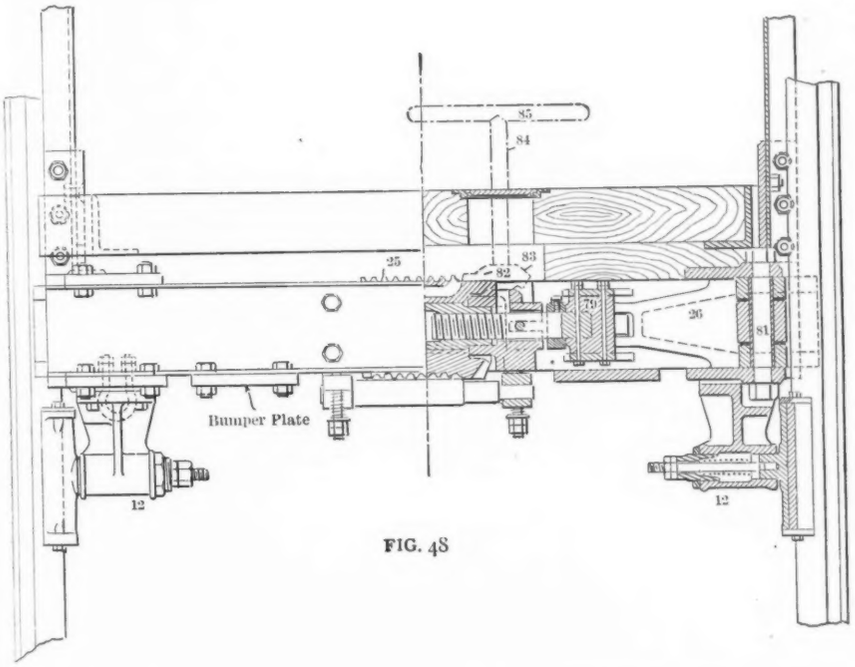


FIG. 48

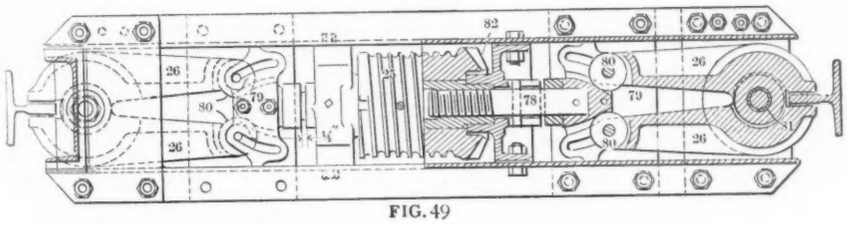


FIG. 49

SIDE VIEW AND PLAN SHOWING CONSTRUCTION OF WEDGE-CLAMP SAFETY

Premium System for Power-house Employees

The Sheboygan (Wis.) Light, Power and Railway Company employs a premium system in the power house to stimulate the employees to greater effort toward economical operation. On a basis of 1.1 cents per kilowatt-hour, which was the average cost per month before the premium system was adopted, 10 per cent. of the total cost incurred for each month below 1.1 cents per kilowatt-hour is divided among the employees, according to rank, the engineers receiving the highest per cent., and the oilers the lowest. The company is paying out in premiums $40 to $80 per month, which indicates a saving in power-house expenses of from $400 to $800 per month.

The general manager of the company stated at the recent convention of the Northwestern Electrical Association that the adoption of the premium system produces an appreciable change for the better in the interest the men take in their work. Every man not only tries to do his best, but also helps the others to do their best. Engineers watch the firemen to see that the coal is economically burned, and the firemen protest if the engineers run the larger units at low load instead of putting in smaller units. Every employee is on the watch for hot bearings, which would mean costly repairs, and carelessness in one employee will not be tolerated by the others.

The International Book and Paper Exposition

The International Book and Paper Exposition, to be held at Paris, France, July 21 to November 1, inclusive, will cover comprehensively the various departments of book and newspaper publishing, as well as the finished product. The American section will include numerous mechanical exhibits, such as typesetting and typecasting devices, presses, paper-cutting machines, etc. The commissioners of the American section are William H. Tolman and Anna C. Tolman, 287 Fourth avenue, New York.

Coal Consumption of Boilers

By N. A. Carle

The burning of the necessary amount of coal for operation of a boiler depends upon the intensity of the draft in the furnace, the amount of air furnished to assist combustion, the form of the grates and the kind and condition of the coal and the method of burning it.

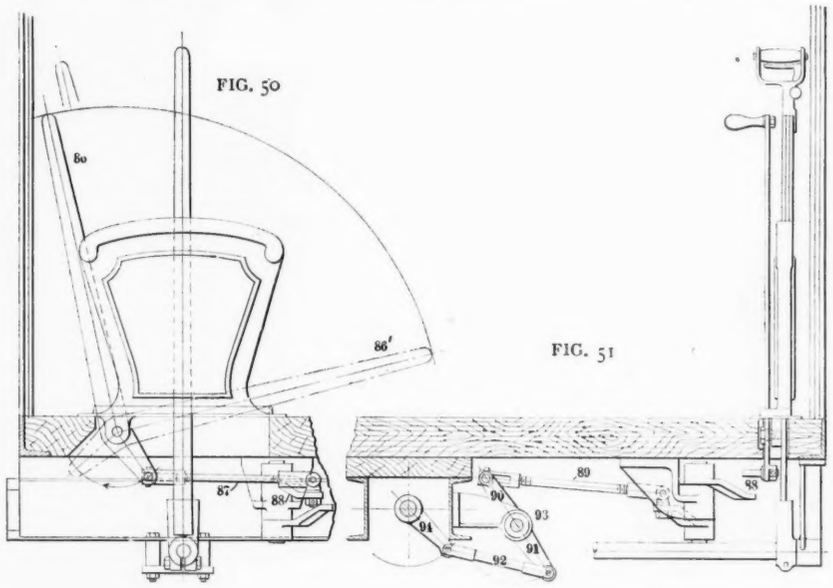


FIG. 50

FIG. 51

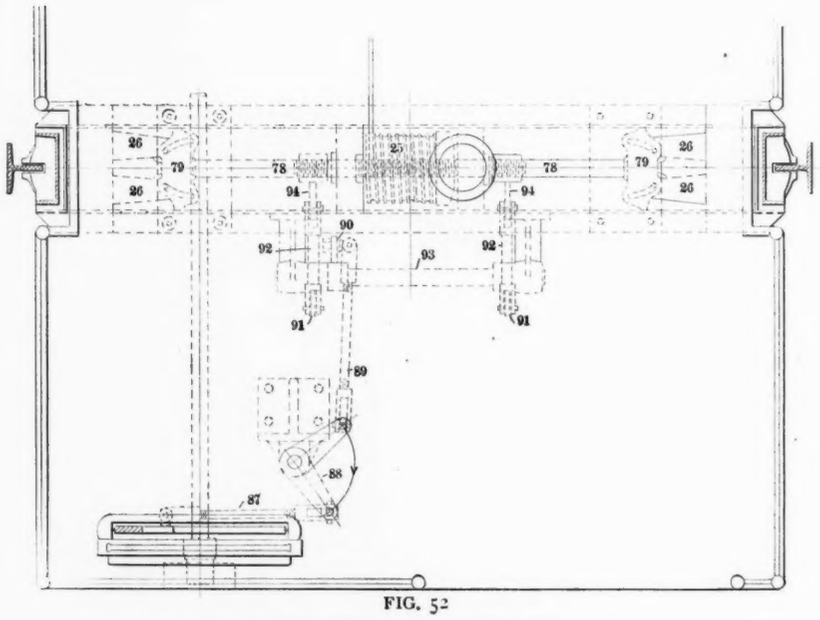


FIG. 52

PLAN AND ELEVATION OF CONTROL MECHANISM

In speaking of the amount of coal consumption under a boiler or the burning capacity of its furnace, it is customary to state this as a definite quantity of coal burned per hour per square foot of grate area. The rate of coal consumption varies directly with the square feet of grate area, the kind and condition of the coal used, the amount of boiler horse-power delivered and the efficiency of the boiler.

The square foot of grate area has a definite ratio to the square feet of heating surface in the boiler depending upon the kind and condition of the coal to be used, and whether the boilers are to be operated with stokers or flat grates. The rate of coal consumption per square foot of grate area usually varies from eight pounds to forty pounds per hour.

The heating value of coal depends upon its chemical constituents and its physical condition of moisture and size of pieces. These variables are usually combined by rating the coal as having a certain number of heat units per pound, which amount is determined by tests in a coal calorimeter and the coal is commercially designated by this name and heating value per pound. This ranges from 10,000 B.t.u. for lignite to values slightly in excess of 15,000 B.t.u. for Georges Creek, New River and Pocohontas coals.

The horse-power delivered governs the amount of coal burned in a direct ratio, except for the variation in the efficiency of the boiler at different per cents of rating, which ranges from 55 per cent. to 75 per cent., with an average value of 65 per cent.

The chart on page 233 is designed to read directly the amount of coal burned per hour per square foot of grate area for boiler horse-powers up to 800, using various kinds of coal, with boilers of grate areas from 50 to 120 square feet and operating from 55 to 75 per cent. efficiency. The superheating of the steam is figured as an increase in horse-power at the rate of 5½ per cent. of the boiler horse-power for each 100 per cent. of superheating.

Example

What would be the rate of coal consumption per hour per square foot of grate area for a 300-horse-power boiler operating at its rating with an efficiency of 65 per cent., and burning coal of 13,000 B.t.u. quality on a grate having an area of 100 square feet?

Starting with 300 "Boiler Horse-power" on the line marked "Without Superheat" read up to 13,000 B.t.u. "Heat Value," then across to 65 per cent. "Boiler Efficiency," then down to 100 square feet "Grate Area," and across to 11.8 pounds of coal per hour per square foot of grate.

Example

What would be the rate of coal consumption per hour per square foot of grate for a 400-horse-power boiler equipped with stokers burning coal of 12,000 B.t.u. quality, with a grate area of 90 square feet, operating with an efficiency of 65 per cent., at 50 per cent. above rating, and furnishing steam superheated to 150 degrees Fahrenheit?

Starting with 600 "Boiler Horse-power" on the line marked "With Superheat," read across to 150 degrees "Superheat," then up to 12,000 B.t.u "Heat Value," then across to 65 per cent. "Boiler Efficiency," then down to 90 square feet "Grate Area" and thence across to 30.8 pounds of coal per hour per square foot of grate area.

The first regenerator steam turbine to be installed in America is being placed at the Irondale steel works of the International Harvester Company at South Chicago. The turbine works upon the system devised by Professor Rateau, the exhaust from engines running reversing and intermittently passing through a regenerator, which consists of a large cylinder inclosing heavy saucers of cast-iron filled with water. These absorb heat from the exhaust steam from the intermittently operating engine and give off the heat with the production of steam under the reduction of temperature while the engines are stopped. As the interruptions are but momentary, sufficient steam is furnished to maintain the turbine in constant operation without fluctuation of pressure.

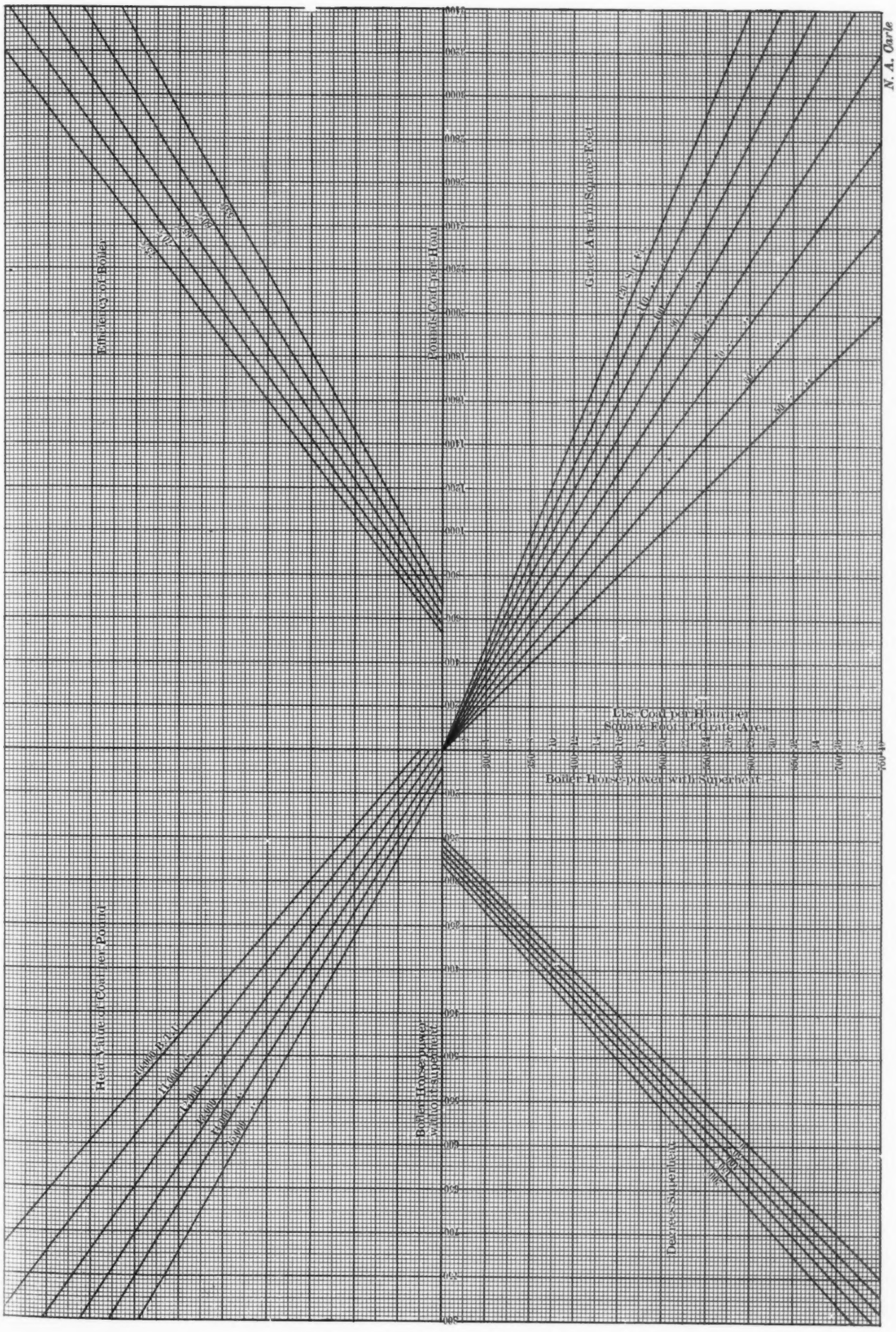

N. A. Carle
Efficiency of Boiler
Pounds Coal per Hour
Grate Area in Square Feet
Lbs. Coal per Hour per Square Foot of Grate Area
Boiler Horse-power with Superheat
Heat Value of Coal per Pound
Boiler Horse-power without Superheat
Degrees Superheat

DIAGRAM FOR DETERMINING AMOUNT OF COAL BURNED PER HOUR PER SQUARE FOOT OF GRATE AREA UNDER A BOILER

Watching a Boiler at Work—An Account of Experiments with a Glass Model of a Water-tube Boiler

By C. Hill Smith

If we take a test-tube filled with water nearly to the top and hold it over a Bunsen flame, the water boils violently and overflows the tube. This violent overboiling is due to the conflicting action of the ascending and descending currents of steam and water in the tube. On the other hand, if we take a tube shaped like a U, the arms of which are connected together at the top, fill it with water and place one leg of the U in the flame, a direct circulation soon commences. The water passes along in one direction and the steam is liberated at the surface. In this case there is very little violent ebullition, because there are no counter currents and the steam is discharged quietly over a liberal surface.

The writer, desiring to ascertain just how nearly a boiler could be designed to work upon the U-tube principle of circulation, produced, after several trials, the model boiler shown on page 235.

This model was built entirely of brass. It contained three drums four inches in diameter and 120 brass tubes one-quarter of an inch in diameter. The tubes were connected into headers and into the circumferences of the drums. The heads of all three drums were made of plate glass, for observation of the interior of the boiler when making steam. It will be noted that the design of the boiler closely resembles the U shape, only that one leg is considerably longer than the other, and there are two legs on the side of the U where the heat is applied.

Each of the three drums serves a special function which will be noted from the description of the experiments. The two legs, instead of being connected together at the top, as was the case in the U-tube, are connected by two separate passages, one for the water to pass through and the other for the steam.

In preparing for the tests the boiler was mounted on a stand, so that the tubes inclined from the horizontal 20 degrees, and the whole was inclosed on all sides by brass plates. Alcohol lamps were placed inside the casing at a point to correspond with the regular location of grates, or at about one-fourth of the distance between the front headers and the rear drum. The steam outlet was located on the rear drum, as was the safety valve, for experimental reasons, although in actual practice the safety valve would be located on the front drum. The feed-pipe was introduced in the rear drum, while the blow-off entered the lowest point of the lower drum, which we will call the mud-drum. The boiler was attached to an open condenser.

The boiler being ready for test, it was filled with cold water until the upper drums were filled to one-half their volume. Candles were placed behind one head of each of the three drums for the purpose of lighting the inside. The alcohol lamps were then lighted and the boiler interior was ready to observe through the glass heads of the drums.

The first action noted was in the front drum, which served as a discharge chamber for all the steaming tubes. The tubes of the lower bank discharged into it through headers, while those of the upper bank discharged into it independently. Many faint, oily-white streamers were seen to rise from the nipples connecting the headers to the front drum, passing upward to the surface of the drum. They resembled little streamers of white smoke. On reaching the surface of the water they passed into the horizontal circulating tubes which connected the two upper drums. These little streamers were heated water, which, being lighter than the water in the drum, rose to the surface. This same action soon appeared from the ends of the upper bank of tubes, the little streamers rising in a similar manner and passing into the horizontal tubes.

By observing the rear drum, the little streamers could be seen coming into this drum from the front drum. Here they turned downward into the vertical tubes which connected the rear drum to the rear headers and the mud-drum. No action could be noted in the mud-drum which fact seemed to indicate that these currents of water passed into the upper tubes and thence into the front drum again, as the action from these tubes appeared very much more decided than the action from the nipples, notwithstanding the fact that the lower tubes were nearer to the flame than the upper ones. This was undoubtedly due to the fact that the heated currents of water remained as near the surface as possible, while the colder water passed to the bottom of the boiler, having greater specific gravity.

Particles of sediment could be seen coming down the vertical circulating tubes into the mud-drum, evidently precipitated from the water that was being heated. This sediment passed to the bottom of the drum, where it remained. A very gradual action was now noted in the mud-drum in the nature of similar currents of water coming down the vertical tubes. These currents acted strangely on entering the drum; they spread out on coming in contact with the colder and denser water lying at the bottom. By placing the finger on the upper portion of the glass head and then on the lower, quite a difference in temperature was noted. Little streamers of heated water soon commenced to pass into the lower bank of steaming tubes which were connected into this drum. They passed across the drum with a sort of shivering motion. A new and very interesting phenomenon was now noticed. Occasionally there would appear from the ends of the steaming tubes little rings of heated water, which shot across the drum with considerable velocity.

The action in the front drum became very much more pronounced and air bubbles appeared from the nipples and tubes. The boiler was circulating water with great rapidity in the same direction and it was noticed, by placing the hand on different parts of the boiler, that all parts were of the same temperature. The air bubbles now discharged in great quantities from the tubes and nipples and rising to the surface disturbed the water level considerably. It was noted that they floated along under the surface of the water before they broke.

Gradually these air bubbles ceased to appear and a new kind took their places. The latter were steam bubbles and they discharged into the drum with greater velocity than the former. On reaching the surface of the water they broke immediately, but they agitated the water level to a much greater extent. Fountains of water would shoot up into the drum for quite a distance and showed very vividly the conditions present in the shell type of boiler, where there are no defined paths for the water and steam to travel and nothing to prevent their conflict with each other. This also shows the cause for wet steam, and the great danger of entraining water with steam, as is the case where the steam is removed from the same space where violent ebullition is present.

While the water level in the front drum was violently agitated, the water level in the rear drum remained perfectly calm. No steam was generated in this drum, as the horizontal tubes connecting it to the front drum only circulated water that had been freed of its steam.

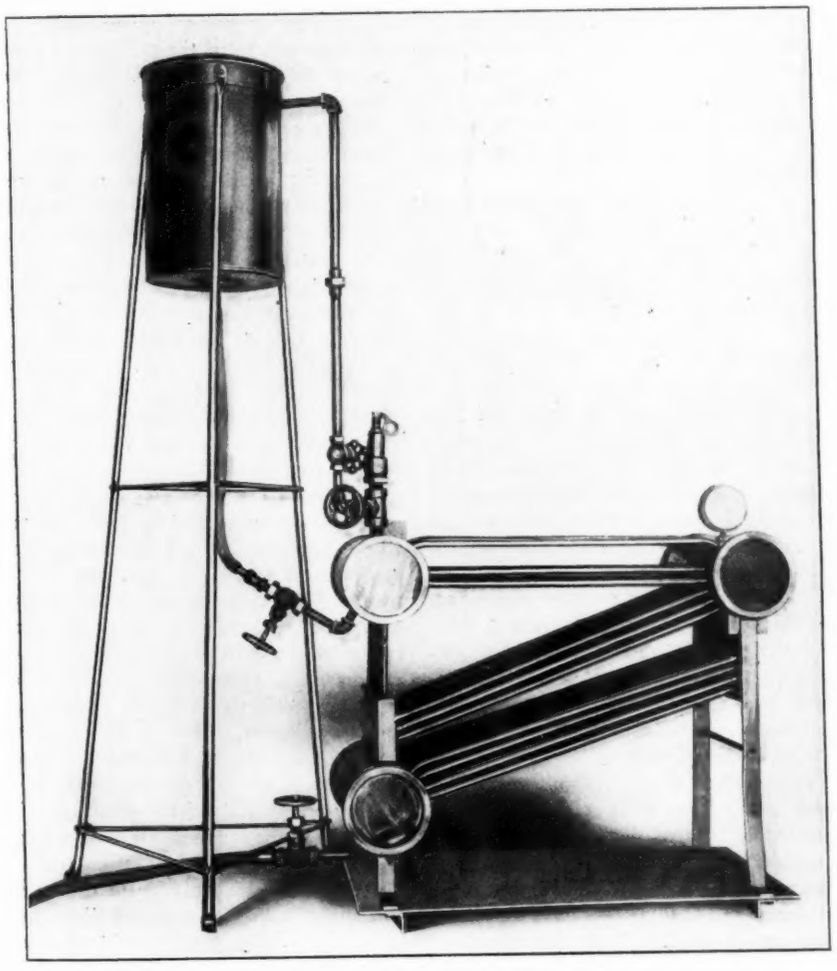
As the steam gage soon registered a pressure of 9 pounds, the main stop-valve was opened to allow the steam to flow to the condenser. The abrupt release of pressure caused the water to expand suddenly and the water level rose about one-quarter of an inch. This was evidently due to the sudden generation of steam caused by the drop in the pressure. This increased ebullition caused a very violent action in the front drum and the circulation of water through the boiler increased greatly in velocity. The nipples and tubes in the front drum discharged great quantities of bubbles. The water level in the rear drum during this increase in ebullition showed but a few ripples, which were evidently due to the vibration of the steam passing into the steam main, or the discharge into the water of the open condenser.

The sudden generation of steam caused

by the opening of the steam valve and subsequent reduction in pressure, it is believed, explains how the partial rupture of the shell of a return-tubular boiler is advanced to a disastrous explosion by the unexpected increased generation of steam due to the lowered pressure.

The steam main was now closed sufficiently to allow the boiler to operate on a constant pressure of about 6 pounds. It operated very smoothly under these conditions and made a very interesting sight with the steam generating in the front drum, where the nipples and tubes discharged great quantities of bubbles.

The action in the mud-drum had in the meantime become well worth watching. In the other two drums the water showed clear in the candle light, but the color of the water in the mud-drum was very murky. Particles of sediment were noted settling to the bottom. The withdrawal of water from this drum by the steaming tubes did not appear to draw this sediment into the tubes, as the drum was of ample size so that the suction was not felt at the bottom where the sediment deposited. This emphasized clearly the advantage of a very large mud-drum to allow of the thorough settling of the sediment.



WORKING MODEL OF A WATER-TUBE BOILER

The condition of the front drum was thought to be too violent for good practice, because the ebullition indicated restriction of circulation. The boiler was put out of operation for the purpose of making changes in this drum to prevent extreme ebullition. The glass heads were removed and other nipples inserted over the nipples that connected the headers into the drum, it being here that the most violent discharge of steam was discernible. These new nipples were cut long enough to reach to the water level, or just a trifle below it.

The glass heads were replaced and the boiler put in operation again. The circulation was similar to that in the first test, and no real difference was noted until the boiler commenced to make steam. Then it was seen that the ebullition in this drum was considerably reduced, the agitation that remained being caused by the discharge from the tubes of the upper bank. This reduction was evidently due to having provided a channel through which the water and steam from the nipples might flow to the surface of the water and so prevent contact with the water in the drum. As the steam and water no longer had to force their way to the surface, the disturbance of the water level was naturally reduced entirely in this direction. The water rose from the nipples in little fountains, the steam disengaging from it in the upper part of the drum.

The boiler was operated under very severe conditions to try the value of this addition of nipples. The main stop-valve was suddenly opened after a considerable steam pressure was obtained. It had very little effect on the water level in this drum, only causing the nipples to discharge fountains of water quite a distance into the drum. No water was thrown into the superheating tubes, as the fountains of water discharged vertically and fell back immediately to the water level.

The value of this attachment being proved, the boiler was blown down and after the water was all withdrawn from the boiler considerable sediment was found in the bottom of the lower or mud-drum. Nowhere else was sediment found, as the drums offered no opportunities for the sediment to settle, being pierced at their lowest points by tubes and nipples. The tubes were inclined 20 degrees, which insured thorough draining of the boiler.

From the foregoing experiments many points of great value for improvement in design of water-tube boilers can be derived. The violent ebullition in the front drum shows conclusively that steam should not be withdrawn from the boiler at a point where ebullition is present, on account of the danger of getting water entrained with the steam. It also shows that any sudden reduction of the pressure causes violent ebullition and priming The front-drum conditions show that this is a good place to locate the safety valve, as the sudden opening of it would cause no liability of priming if the steam is not withdrawn from this drum.

The total lack of any ebullition in the rear drum shows that this is an ideal spot to remove the steam. It was noted that, owing to the large amount of separating surface provided, the opening of the steam valve caused no priming in this drum. Another feature to be noted is the value of a large mud-drum to provide ample opportunity for the sediment to settle, and also to provide a large supply of water for the bottom tubes. It would be impossible to force the boiler hard enough to drain this drum of water, so the danger of burning out these tubes is eliminated.

The provision of the long nipples in the front drum proved the advantage of providing separate passages to allow the steam and water to reach the surface of the water, thus obviating the necessity of their forcing their way to the surface through the large body of water in this drum and so cause violent ebullition.

Absorption Refrigerating Machines

Showing How a Plant is Laid Out, and Giving Practical Directions for Running either a Dry or a Brine System

BY JAMES W. COOPER

The generator of an absorption machine is used for the purpose of separating ammonia from water. This is done in the following manner: Water and ammonia are mixed until they stand by hydrometer test at 0.74 water and 0.26 ammonia at 60 degrees Fahrenheit. This is pumped into the generator in sufficient quantities to cover the steam coils, generally from two inches to four inches. As soon as steam is turned into the coils the ammonia begins to leave the water and rise above it, and if the generator were used in the same manner as a regular boiler the ammonia would serve as the expansion or working gas, instead of steam, until all the ammonia had evaporated.

But instead of using it as a boiler, it is designed to make ice, so the ammonia is passed into what is known as an "analyzer." This analyzer is a series of plates so arranged that the gas leaving the generator will come in contact with ammonia and water coming from the ammonia pump. When the liquor coming from the pump is supposed to be saturated with ammonia, that is, the water has all the ammonia it will hold at that temperature and pressure, the liquor gives up part of its ammonia. Also, the gas leaving the generator is saturated with water, so it gives up part of this water to be returned to the generator, making the analyzer answer a twofold purpose and thereby increase the efficiency of the machine.

Now, in the distillation process it is not possible to get dry gas—that is, pure ammonia—from the generator, so there is used what is called a rectifier. This is simply a coil of pipe with a steam loop to return the condensed liquor to the generator. Water is run over the rectifier to keep it cool, and the engineer soon learns just how cool to keep it. The rectifier takes the gas direct from the analyzer, separates what water is left in the gas, returns the water through the loop to the analyzer and sends the ammonia gas to the condenser, there to be cooled until it becomes liquid. As soon as it becomes liquid it falls to the bottom of the condenser, which serves as the anhydrous receiver, and from here it goes to the expansion valve to be expanded into a gas.

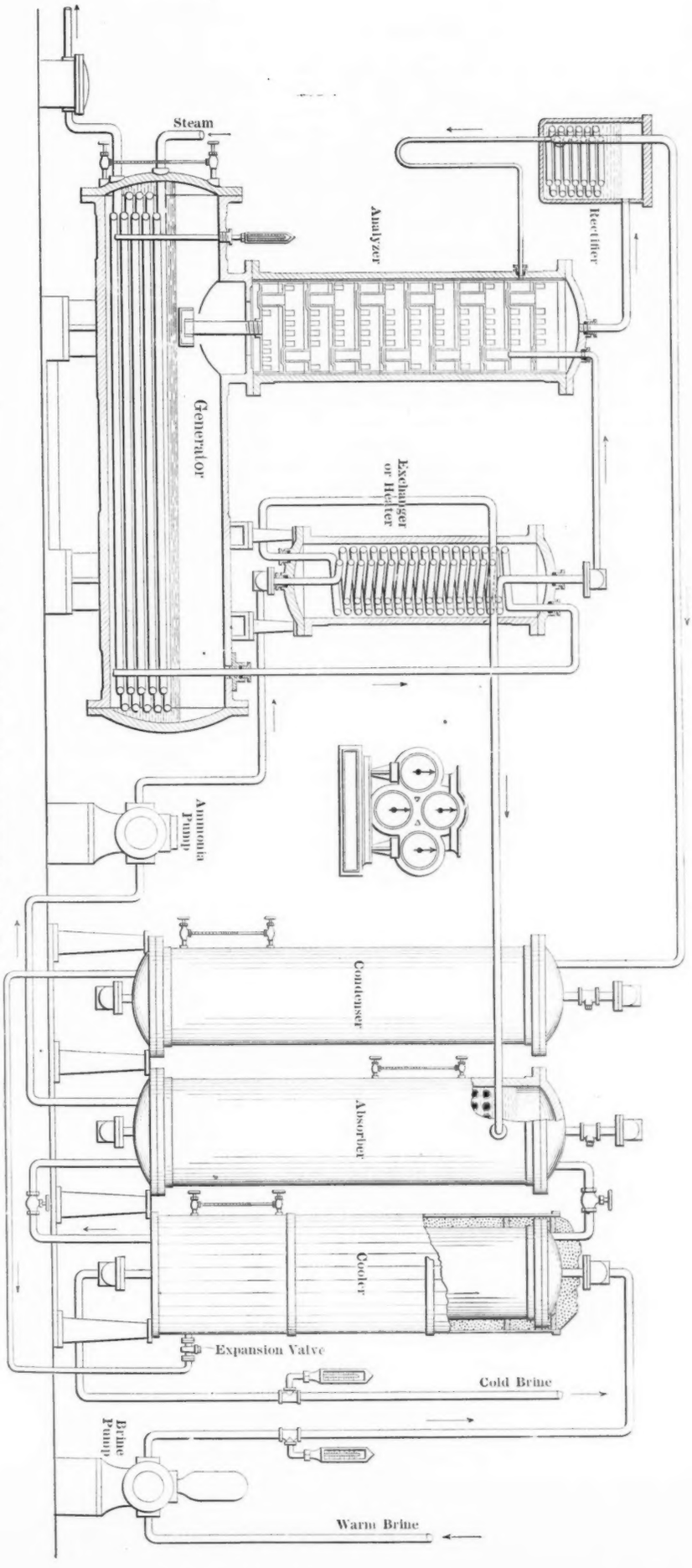
This last is one of the most important functions of the whole process. Ammonia as a gas should never be allowed back of the expansion valve; in other words, keep the pipe leading from the receiver to the expansion valve full of liquor. The reason for this is that ammonia gas will impart heat, while ammonia passing from the liquid to the gaseous state will absorb heat. The anhydrous ammonia should therefore come to the expansion valve as a liquid and begin its expansion at the expansion valve. This holds good whether the direct-expansion or brine system is used. To understand this better the engineer should study the "ammonia scale," or the table of physical properties of ammonia.

Having followed the ammonia gas from the generator to the expansion valve, the next thing to learn is how to make the ammonia do its work and still retain the liquid so as to be able to work it over again. This is done by so arranging the machine that the water left in the generator can be mixed again with the ammonia that was separated from it. This is achieved in the absorber. Absorbers can be of different kinds, but they are all worked on the same principle. The kinds the writer is most familiar with are the "immersed," the "dry" and the "pipe-within-a-pipe." The difference between the immersed and the dry lies in the fact that the former is kept very nearly filled with rich liquor, while the latter is run as nearly empty as possible. The "pipe-within-a-pipe" is simply one pipe running into another, each coming from opposite parts of the machine.

The absorber operates as follows: The "weak liquor," as it is called, is taken from the bottom of the generator, passes through what is known as the "exchanger" (generally called a heater or economizer), and from this goes either to a weak-liquor cooler or directly to the absorber. In the absorber it is sprayed so as to pass over the gas coming from the chill rooms, or if the brine system is employed the two mix and form what is known as a "rich" liquor. In all ice machines, except open-air condensers, water has to be pumped over the condensers, rectifiers and absorbers. This water does not come in contact with the ammonia that is in the pipes. The real process by which heat is forced out of the system takes place in the condensers, where the ammonia changes from gas to liquid. It should be remembered that there is no vacuum on the condenser, but a pressure that corresponds with the temperature at which the gas liquefies. This liquid is called anhydrous ammonia, and its boiling point, if pure, is 28½ degrees Fahrenheit below zero at atmospheric pressure.

In the absorber there are two heat quantities to be taken care of; one is called the "heat of absorption," and the other is the difference in the temperature of the weak liquor after it passes the exchanger and the temperature of the rich liquor returning to the generator through the pump. Of these the heat of absorption is considerably greater, nearly equaling the latent heat of steam at atmospheric pressure, and the secret lies in so regulating the weak-liquor valve that it will absorb the largest amount of ammonia with the lowest pressure possible. In working a machine the engineer will have to determine this by practice; sometimes increasing the pressure by slowing up on the weak liquor gives good results, while at other times increasing the flow of weak liquor will give better results. A few hours' experience and an intelligent engineer will master this all right.

To state it more briefly the absorber is simply a cylinder, with pipes running through it and through these pipes water is circulated to cool the mixture. On the inside of this cylinder gas and weak liquor are mixed together, so as to form rich liquor, which is to be returned to the generator to be reseparated and used over and over. The weak-liquor exchanger is merely a heater, the rich liquor passing through coils of pipe or a shell on its way to the generator, and the weak liquor passing on its way to the absorber. The principle is the same as feeding through a heater to a boiler and using the exhaust to heat with, only in the case of the exchanger both elements are liquid, one hot and the other cold. With the brine system it is necessary to have a cooler in which the required temperature is obtained. The cooler itself is similar to the absorber, being a cylinder with coils of pipe running through it for the brine to be pumped through. The ammonia evaporates over the pipes to cool the brine, although it makes no difference whether the ammonia goes through the pipes or over them. The gas leaves the generator, passes through the analyzer to the rectifier, thence to the condenser, where it changes to a liquid; from the condenser to the receiver, then through the expansion valve to the cooler. From the expansion valve it is allowed to expand into a gas again, and during this expanding from a liquid to a gas it absorbs its heat. A large



DIAGRAMMATIC ELEVATION OF AN ABSORPTION REFRIGERATING SYSTEM

pipe leads from the top of the cooler to the top of the absorber, through which the gas passes. Once in the absorber the weak liquor mixes with it, the two become rich liquor, and are then returned to the generator to be worked over again.

Starting up a Brine Plant

Let us suppose one is called upon to start up a new plant where the brine system is used. The builders will give this plant a hydrostatic test of three hundred pounds gage pressure, and report everything in good order. If salt is to be used simply mix salt and water together until the salinometer registers 10, which will allow one to operate to within three to five degrees above zero Fahrenheit without freezing. But in cold-storage work it is desired to go considerably below zero, so chloride of calcium is used to make the brine with. It is mixed with water and thoroughly agitated until it becomes of the required density; it is generally carried to about 12.50 heavy liquid hydrometer test. This will permit running as low as thirty degrees below zero, without freezing. The peculiarity of calcium brine is that if too thick it will congeal at a much higher temperature, the same as if it were too thin, so it is better to have it just right. (This is another scale the engineer should become familiar with.)

Now to start the machine. As soon as the hydrostatic test is completed draw all the water from the machine, fill the generator with aqua ammonia (which is simply water and ammonia mixed together) until the steam coils are sufficiently covered, and turn the steam onto the generator. As soon as the ammonia begins to get hot the generator pressure will begin to rise, showing that it is time to start the water or circulating pump. See that the water is on the condensers, rectifiers and absorbers. Now start the brine pump, at the same time getting the ammonia pump ready. Look at the receiver glass to see how much anhydrous ammonia there is; if enough, open the expansion valve slightly, and then open the valve on the gas line between the cooler and the absorber. Next open the weak-liquor valve and start the ammonia pump. If you find there is not enough pressure, you will have to strengthen the "charge," as it is called, with anhydrous ammonia.

This is a very simple process. Shut the expansion valve, convey a hose or pipe (if hose, it must be a stout one) to the cooler and the anhydrous cylinder, open the valves cautiously and let the anhydrous ammonia flow in. Keep this up until you have the desired amount, then open the expansion valve and try again. You will be able to tell what the machine is doing by the difference in the thermometer connected on the brine pipe at opposite ends of the cooler. This is called the "range." Suppose the brine is coming from the house at ten degrees and going to the house at four degrees,

this is a range of six, and we generally say "she is doing six," or whatever it is. The pressure that should be kept on the freezing side of a machine depends upon the temperature of the brine. The water tries to keep from five to ten pounds above the brine if we are running near zero; if higher than that it is desirable to keep them even. Of course, the absorber pressure should always be lower than the cooler.

Foreign gases get into an ice machine and have to be blown out; also the weak liquor will get into the cooler and have to be purged out. For blowing the foreign gases out there is a pipe on the absorber; simply work the machine fast for a while until the foreign gases are forced over to the absorber, then burn or blow them out, being careful not to lose any more ammonia than can be helped. In purging out the weak liquor from the cooler, shut the gas line running from the top of the cooler to the top of the absorber, open the purge line on the bottom of the cooler, let all the weak stuff blow over to the absorber and then pump it back to the generator. When the cooler is dry, shut the purge line and open the gas line. In testing for leaks use litmus paper, sulphuric acid or a sulphur candle. If litmus paper is used wet the paper, hold it near the pipe and if ammonia touches the white paper it will turn the paper purple. This paper can be dried and used again. If sulphuric acid is used, hold the bottle near the suspected leak and watch the color of the vapor that arises. If a sulphur candle is employed light it and hold it near the supposed leak. Leaks should be stopped quickly as they are expensive.

After an absorption machine has been run for quite a while (how long, the engineer will have to determine by the action of the machine), the cooler will have to be blown out. This is done in the following manner: There is a pipe line running from the bottom of the cooler to the absorber, called the purge line; sometimes this line runs direct to the bottom of the absorber and sometimes from the bottom of the cooler to the gas line previously described. Open the valve at the bottom of the cooler and let all the weak liquor escape to the absorber where it will be pumped back to the generator. After purging shut the purge line and go on with the machine as usual. The first indication that the cooler needs purging is that the ranges of temperature become less; that is, a lower degree of temperature between the outgoing and the incoming brine is manifest. Say the incoming brine has been running at ten degrees below zero and the outgoing at four degrees; this would give a range of six degrees. Now, say the temperature of the outgoing brine rises so the thermometer shows only six degrees. First look over the machine to see if everything is as it should be; if so, go to the gage glass on the cooler, open it to see if it shows a watery-looking substance, in which case it needs purging.

Priming or "puking" is the worst thing an engineer will have to contend with in an absorption machine. This consists of throwing all the aqua ammonia into the condensers and leaving the generator without any liquid in it. The first indication that this has occurred is that the ice melts off of the expansion-valve air pipe. If the engineer notices this, let him go to the generator at once and see how the liquid stands in it. If it is extremely low, very likely priming has taken place, but to be sure of this look at the pressure gage on the cooler. If the pressure begins to drop fast then priming is a certainty; in less than five minutes the gage, which is always a compound one, going from five pounds pressure to 28 inches vacuum. If there is a pipe running from the receiver to the absorber the difficulty is easily adjusted; if not, it means hard work.

Let us assume that there is no pipe from the receiver to the absorber, then all the weak liquor must go through the cooler back to the absorber, thence to the generator. Keep the ammonia pump working at its best to get as good a vacuum on the absorber as possible. Next open the expansion valve so as to get all the weak liquor out of the receiver and condenser into the cooler, and if the pressure is still below that of the absorber and they both show a vacuum at this time, simply shut the expansion valve and open the anhydrous charging valve. This will let the air run in from the outside and cause the cooler to show atmospheric pressure, which will be greater than the pressure in the absorber and the liquid will go to the absorber and be pumped to the generator again. Keep repeating this operation until the machine is normal. The cause of this condition may be that the charge is too weak or the machine is working too fast and the generator is very dirty. Of course, this weak liquor will have to go through the purge line at the bottom of the cooler and to keep a greater pressure on the cooler than on the absorber the gas line will have to be closed between the cooler and the absorber. This will force the liquid out faster.

Features of the Western Electric Company's New Plant

The Western Electric Company has published a 24-page booklet descriptive of its recently acquired 110-acre plant at Hawthorne, a suburb of Chicago, Ill. It is entitled "Hawthorne Works." Certain features of the new plant possess especial interest from an industrial standpoint, such, for instance, as the plan of storing coal under water, an idea adopted from the British admiralty. Two storage bins of 4000 and 10,000 tons capacity are located below the normal ground level. Coal can be dumped into them and taken out by means of a locomotive crane fitted with a grab-bucket. The bins are kept flooded with water, thereby eliminating danger of spontaneous combustion. It having been shown by tests conducted in Europe that nearly 30 per cent. of the heating value of coal is lost in six weeks when exposed to the air, the company expects to reduce the losses of coal through the flooding process to approximately two per cent. of its heating value, after six to twelve months' storage. By this system enough coal can be stored at one time to operate the plant, under normal winter conditions, for four months. This method of storing coal is comparatively new in this country.

The method of handling coal and ashes is noteworthy also. The property is divided by railroad tracks of such grade that cars may be switched over an elevated spur track into the storage bins above the boiler-room. The bins are of 1000 tons capacity and discharge through spouts into a hopper scale mounted on a truck. There is a hopper scale for each division of boilers.

Ashes are discharged at the end of a chain-grate stoker into a bin of one and a half tons capacity located below each boiler. From these bins the ashes may be delivered into gondola cars and taken over a depressed railroad track to various places about the plant where filling-in is required. The boiler-room has an area of of 20,700 square feet and is designed for two chimneys, each 12 feet inside and 22 feet outside diameter and 250 feet in hight above the floor. There is provision for two batteries of eight boilers each for continuous service of 3000 kilowatts, and for two-hour service of 4500 kilowatts.

The various buildings are connected with the power house by well lighted and ventilated tunnels of uniform hight of seven feet. All the machinery in the entire plant is electrically driven, an individual-drive system being employed, except in a few cases.

A crematory having approximately an area of 1600 square feet cares for the shavings and chips from the carpenter shops.

For fire protection there is a reservoir of 5,000,000 gallons capacity; also a water tower containing six steel tanks with a total capacity of 213,000 gallons, connected with an automatic sprinkler system which has been installed in all but the foundry, forge shop and main body of the machine shop.

Forty-five hydrants, each with three or four independent gates, are located on the property, connected with the mains that supply the sprinkler system. In a fire house adjoining the tower are two 1500-gallon Underwriter fire pumps.

Catechism of Electricity—XXI

ELECTRICAL MEASUREMENT OF RESISTANCES

452. *If change of temperature affects resistances as has previously been stated, is it not necessary in very accurate resistance measurements made by means of the Wheatstone bridge to allow for the effect which the temperature at the time of the measurement has upon the bridge resistances?*

Ordinarily, a change of temperature affects resistance to the extent of about four-tenths of one per cent. per degree

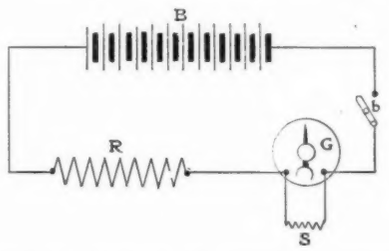


FIG. 99. DIAGRAM OF DIRECT-DEFLECTION METHOD OF MEASURING RESISTANCES

Centigrade of change, as has already been explained. In the Wheatstone bridge, however, the resistance coils are of either german silver or platinoid, or sometimes manganin, these materials being used largely on account of their comparative low temperature coefficients or change in resistance per degree change in temperature. For example, the temperature coefficient of german silver is only about 0.00044, but unless the temperature at the time the resistance measurements are taken is the same as that at which the resistances of the Wheatstone bridge have been calibrated, which latter temperature is stamped on the bridge, a correction should be made if it be desired to secure extremely accurate results.

The correction, however, need be applied only to the known resistance r, because the resistances a and b being of the same kind of metal the ratio of a to b as used in the formula $x = \frac{b}{a} \times r$ will not be affected by any change in temperature.

453. *Determine very accurately the resistance introduced in the x arm of a Wheatstone bridge composed of german silver resistances if a balance be obtained with $a = 100$, $b = 1$ and $r = 542$, the temperature at which the bridge resistances were calibrated being 19 degrees Centigrade, and the temperature when the balance was secured being 15 degrees Centigrade.*

For 19 — 15, or 4 degrees, change in temperature there results, according to the temperature coefficient of german silver given in answer to Question 452, a correction factor of $4 \times 0.00044 = 0.00176$. As the balance was secured at a lower temperature than that at which the bridge resistances were calibrated, their exact resistance would be less than the figures marked above them on the bridge, because the resistance of a metal decreases with a decrease in temperature; it therefore becomes necessary to multiply the recorded value of the known resistance, or 542 ohms, by 1/0.00176, or 0.99824, which gives 541 ohms for r. Proceeding as usual to find the unknown resistance from this value of r, we have $x = \frac{1}{100} \times 541 = 5.410$ ohms.

454. *What method is used for measuring resistances higher than those which can be ascertained by means of the Wheatstone bridge?*

The direct-deflection method.

455. *Illustrate and describe the direct-deflection method.*

Suppose a battery B of many cells be arranged in series, as in Fig. 99, with a delicate high-resistance d'Arsonval galvanometer G, a high resistance R of known value, and a shunt S composed of three coils, one of a resistance of $\frac{1}{9}$, the second $\frac{1}{99}$, and the third $\frac{1}{999}$ that of the galvanometer. Before commencing the test it is necessary to have the number of cells in the battery, the value of the galvanometer shunt used, and the position of the control magnet of the galvanometer such that upon closing the key b a scale deflection of convenient extent is given by the galvanometer. Suppose the deflection obtained with the resistance R in circuit be called D.

If, then, in place of the resistance R, the unknown high resistance X be introduced in circuit, a deflection of a certain number of scale divisions which may be represented by D_x will be given by the galvanometer upon closing the key b. The value of the resistance X may then be found from the formula

$$X = R \frac{D}{D_x}.$$

456. *What would be the resistance of circuit by the method described in answer to Question 455 if the galvanometer gave a deflection of 100 scale divisions with a known resistance of 1,000,000 ohms, and a deflection of 25 divisions with the unknown resistance in circuit?*

Substituting for R, D, and D_x, their respective values of 1,000,000, 100, and 25 in the formula $X = R \frac{D}{D_x}$ there results

$$X = 1{,}000{,}000 \times \frac{100}{25} = 4{,}000{,}000 \text{ ohms.}$$

The resistance of the body is therefore 4 megohms.

457. *For what class of work is the direct-deflection method of measuring resistances suitable?*

For the measurement of resistances greater than one megohm (1,000,000 ohms). The insulation on wires and cables and insulators for low potential work are usually tested by this method.

458. *When insulators are tested by the direct-deflection method how are they arranged?*

They are usually placed, inverted, in a box lined with zinc, as shown at z, Fig. 100. The box is partly filled with water, and water is also poured into the insulators i and i in sufficient quantities to reach within an inch of the rims. The rims should be dry, and the test should not be made on a damp day. One pole of the battery B is connected with the zinc lining of the box and the other pole is connected to one terminal of the galvanometer G, connection being made on the shunt box S. The other terminal of the galvanometer is then connected to one end of an insulated wire a, this connection being also made on the shunt box S. There should now be no deflection in the galvanometer if the wire a is perfectly insulated. To insure this the wire had better not

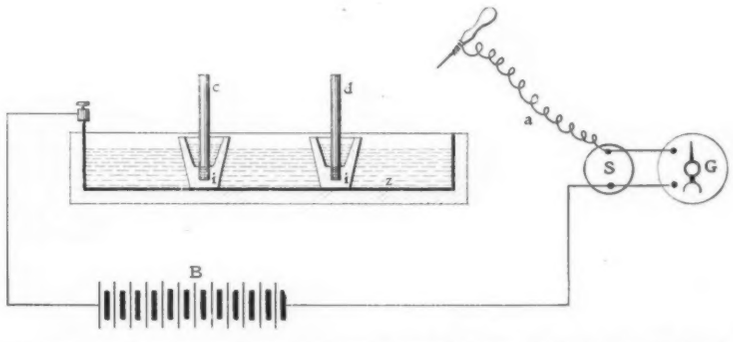


FIG. 100. METHOD OF TESTING THE INSULATION RESISTANCE OF INSULATORS

come in contact with the ground or wall of the room. It should be an "air line" as nearly as possible, and supported, if necessary, only by wood or rubber. The insulation of the testing wire being thus rendered certain, the free end of this wire, supported by a wooden holder, should be touched to the metal pins c and d inserted in the sockets of the insulators which are being tested, the galvanometer being shunted with the $\frac{1}{999}$ shunt. In cases where there is no reading, the insulation has stood the test. Such cases should then be tried with the $\frac{1}{99}$ shunt; then, if this test causes no deflection, with the $\frac{1}{9}$ shunt, and finally, without any shunt whatever.

459. *Is not the direct-deflection method suitable for testing insulators intended for high-potential work?*

No; because insulation for high potentials should be tested not so much for simple resistance as for its ability to withstand high potentials; again, the resistance when subjected to a low potential may be quite different from that under a high potential on account of the electrostatic attraction between the particles, bringing them closer together. In testing such insulators at the factory, it is common to subject all parts to a difference of potential several times that which they are to stand in use.

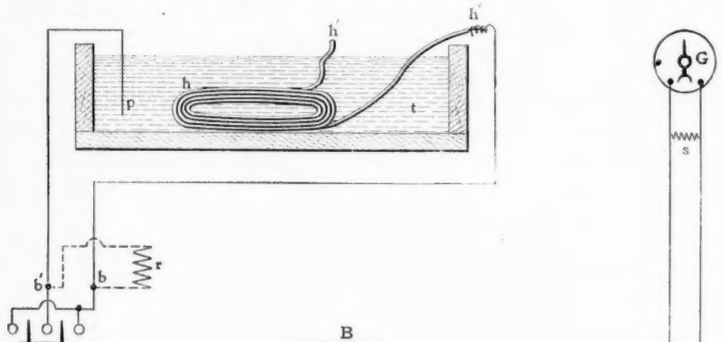


FIG. 101. METHOD OF TESTING THE INSULATION RESISTANCE OF A WIRE

460. *When the insulation resistance of wires is tested by the direct-deflection method, what is the course of procedure?*

The insulated wire is first immersed in a tank of water for a period of forty-eight hours, in order to allow the insulation to become thoroughly saturated, after which connections are made with the wire and the water in the tank, as shown in Fig. 101. The positive terminal of the battery, *B*, is connected to the contact blade *m* of the battery-reversing key shown at *m l*. The negative terminal of the battery is connected to one of the galvanometer terminals, and the other terminal of the galvanometer is connected to the contact blade *l*, of the key. A known resistance *r* has its terminals connected to *b'* and *b* of the key. This connection is shown with dotted lines. The shunt for regulating the sensibility of the galvanometer is indicated by *s*. The circuit runs from the positive terminal of the battery to *m*, and when the key *m l* is thrown to the left, the circuit continues to *b*, through the resistance *r*, back to the binding post *b'*, and thence to the binding post *l*. The circuit then continues on through the galvanometer *G* and back to the negative terminal of the battery. On completing this circuit a certain deflection will be obtained at the galvanometer, and its amount should be recorded.

The coil to be tested is shown at *h*, and is immersed in a tank *t*, filled with water. The terminals *h'* and *h''* are brought over the edge of the tank. The terminal *h''* is cleaned of insulation and connected to the binding post *b*. The end *h'* is allowed to hang free, but out of water. From the binding post *b'* a wire is run into the water in the tank, the end *p* of this wire being cleaned so as to obtain a good electrical contact with the water. The terminals of the known resistance *r* are then disconnected from the circuit. The key is thrown to the left,* closing the circuit from the positive side of the battery *B* to *m* and to *b* as before; to the wire terminal *h''*, through the insulation of the coil *h*, through the water to *p*, thence back to *b'*, through *l* to and through the galvanometer *G* and back to the negative terminal of the battery. A deflection is obtained in the galvanometer *G* on completing this circuit. By comparing this deflection with the one obtained through the resistance *r*, the insulation of the coil *h* can be calculated as described in answer to Question 455.

By reversing the key *m l*, Fig. 101, the circuit will run from the positive terminal of the battery *B* to *b'*, thence to *p*, through the insulation to *b*, to *l*, through the galvanometer *G* and back to the negative terminal of the battery. The first position of the key connects the positive terminal of the battery to wire, and the negative terminal to the ground; the second position connects the negative terminal of the battery to wire and the positive side to ground. The deflections resulting, however, should be the same in both cases.

*The key is actual, made to be depressed instead of moved sidewise, and, when released, the blades rise. Depressing the actual key effects the same results as throwing the diagrammatic key to the left.

461. *When the insulation resistance of cables is tested by the direct-deflection method, what is the course of procedure?*

Cables are tested in the same manner as wire, each conductor being tested separately. When the cable is lead-covered, however, it is not immersed in water. In testing such a cable the conductors are all bound together at one end and then to the lead sheath, with copper wire. At the other end of the cable the conductors are separated from each other. As each conductor is tested, it is separated from the rest and connected to the test wire. When the test on it is finished it is bent back in its original position and the next wire taken.

In Fig. 102, *a* is the lead sheath of a cable and *w* its interior insulation. At *h* and *h'* are the conductors, each with its individual covering of insulation. At *h* the ends are bared and bound together with the wire *d*, which is then bound around the sheath *a*. The conductor being tested is connected at *z* to the test wire *b*, which leads to the post *b*, Fig. 101. At *e* a conductor *b'* is connected to the binding wire and leads to the post *b*, Fig. 101. The test circuit is thus complete from *b* to the conductor at *z*, thence through the insulation, to the lead sheath, to the binding wire and back to *b'*. The conductors at the end *h'* of the cable, being separated, prevent short-circuits, that is, electrical contacts between themselves.

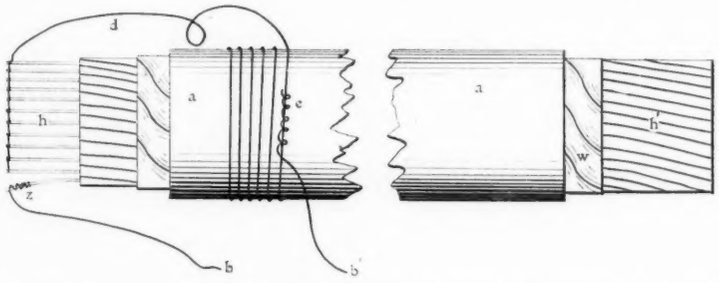


FIG. 102. CONNECTIONS FOR TESTING THE INSULATION RESISTANCE OF A CABLE

462. *Are there any other methods in common use for measuring resistances?*

There is one other method very often employed for measuring comparatively low resistances. It is called the fall of potential or drop method and is based on Ohm's law. If the fall of potential or the pressure in volts across a certain resistance, and the current in amperes flowing through it, be both measured simultaneously, the resistance in ohms will be equal to the volts divided by the amperes. The instruments for measuring the volts and the amperes are known respectively as voltmeters and ammeters; these instruments will be discussed later on. Representing the volts by *V*, the amperes by *I* and the resistance by *R*, the formula is

$$V \div I = R.$$

463. *What is the value of a resistance across which a voltmeter indicates 16 volts while the current passing through it, according to the ammeter, is 4 amperes?*

Substituting for V its value of 16, and for I its value of 4 in the formula, there results $R = \frac{16}{4} = 4$ ohms; this is the value of the resistance.

464. *Are any special precautions necessary in applying the drop method of measuring resistances?*

When the resistance of a highly inductive circuit, as that of an armature of a generator, or the windings of a transformer, is to be measured by the drop method, the voltmeter must be disconnected before opening the main circuit; otherwise, the inductive discharge, which is in the opposite direction to the main current, will cause a reverse deflection and probably bend the pointer of the meter.

465. *If the insulation resistance of a certain length of wire or cable be determined by measurement, can its insulation resistance per mile be calculated from this?*

Yes, if l = length in feet of the wire or cable tested; r = its insulation resistance as determined by measurement; and R = its resistance per mile; then

$$R = \frac{lr}{5280}.$$

466. *What is the insulation resistance per mile of a cable 2640 feet long if its insulation resistance measures 1200 megohms?*

Substituting for l and r their respective values in the formula

$$R = \frac{lr}{5280},$$

there results

$$R = 2640 \times \frac{1200}{5280} = 600 \text{ megohms};$$

this is the insulation resistance per mile.

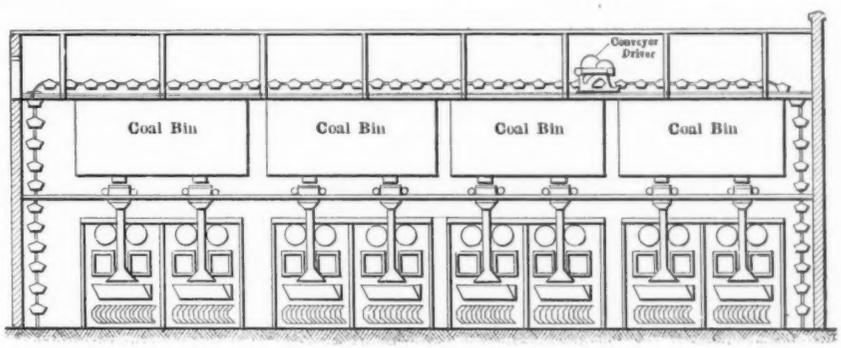
Engineers' Civil Service Examination

The United States Civil Service Commission announces an examination on April 10 to secure eligibles for the position of engineman, at $900 per annum, in the Quartermaster's Department at Large, Fort H. G. Wright, New York. Among the qualifications letter-writing will weigh as 10, practical questions as 65, and experience as 25. Applicants must show in their applications that they have a thorough knowledge of and experience with boilers, engines, steam pumps, dynamos, switchboards, and storage batteries, and ability to read and test wattmeters, and to care for and make slight repairs to arc lamps, pole lines, house wiring and electric-lighting fixtures.

Water-tube versus Horizontal-tubular Boilers and Hand versus Mechanical Firing

By Ed. H. Lane.

I was very much interested in the article on the Wood Worsted Mill in February Power, and would like to make a few comments on it. I am not familiar with the class of work which this plant has to do, but I suppose the boilers were put in for the same purpose ours were, viz., to generate steam; although at first glance one would think that it was to see how much ground they would cover. The engine equipment looks to be first class, but with "the engine serving in effect as a reducing valve" for the heating system, there is undoubtedly a whole lot of money tied up in the low-pressure end of the engines, in the way of large, heavy piping, low-pressure cylinders, condensers and auxiliaries, which certainly will not pay the customary rates of depreciation usually charged up to the engine end of a steam plant.



SUGGESTION FOR A COAL-CONVEYER SYSTEM

The really interesting part of the article, however, is in the description of the boiler-room. I am willing to admit that the old-style return-tubular boiler is not by any means a "back number" for small plants, and that a coal-conveyer system for a plant like this would not pay, but with a few changes in the general equipment of the boiler-room I think the owners of the Wood factory would be a few dollars ahead. Suppose that it had been decided to use standard water-tube boilers instead of the return-tubular type. By placing them in two rows, each row facing the other, the boiler-room would need to be about 125 feet long, only, instead of 328 feet.

To install coal hoppers in this building would call for a higher roof, of course, the alteration required being in the walls at the ends and one side; the engine-room wall is high enough. On the other hand, it would save 200 feet of roof, 200 feet of side wall and 200 feet of foundation; and the last named, by the way, is no small item, as anyone who has built foundations along a river bank can testify. On the whole, I believe a water-tube boiler house can be built a great deal more cheaply than that described in the article mentioned.

I doubt if it will be claimed that the return-tubular boiler is quite as efficient as the water-tube type, neither has it the overload capacity. Likewise, the stoker is, I think, conceded to furnish the most efficient way to burn coal, and a lower grade of fuel can be burned with stokers, so I am of the opinion that most builders of modern plants are justified in installing them.

The boilers in the Wood plant are strictly first-class articles, to judge from the description, and are not to be picked up in any boiler maker's back yard. They evidently did not cost a great deal less than the same horse-power in water-tube boilers would have cost. In fact, if the life of the boiler is taken into consideration, they will cost about the same. Nevertheless, there are water-tube boilers here in Kansas City which were put into service in 1887 and are still doing business. The steam line and fittings will probably cost more per horse-power in the return-tubular plant.

At one of the large "weinerwurst" factories of this "village" the "Armstrong" system of handling ashes is in vogue, and each man has to wheel the ashes of 1000 horse-power of boilers 50 feet. Negroes are employed, and it is generally conceded (especially by the police) that the Kansas City negro is a good specimen of animal muscle. There are forty boilers and it requires eight men each watch, when the boilers are all in operation, to handle the ashes. With a conveyer system one man could handle the ashes for the 8000 horse-power of boilers, for both watches, using the automatic loader, of course.

For handling coal with the conveyer system, dump cars are usually employed, which empty the coal into one main hopper, whence it goes by gravity to the buckets; and it requires only four men to handle the coal, for both watches, for 8000 horse-power of boilers. For handling the coal by the "Armstrong" system five men are allowed for twenty boilers, so for forty boilers ten men would be required.

Now for a few facts and figures concerning firemen, water-tenders, etc.: Forty boilers means eighty doors, and at the rate of five doors the man it would require sixteen firemen each watch. One man could not tend the water for all the boilers; it would require at least two men, so for both watches there would be needed thirty-two firemen and four water-tenders. *Per contra,* the water-tube plant would require four firemen and one water-tender for each watch, or eight firemen and two water-tenders all told.

Twenty days should be the limit to run a return-tubular boiler without washing out, so that would mean two boilers to be washed out each day. As the tubes must be swabbed, red-hot ashes taken out and wheeled about half a mile, the scale removed from the boiler, and the furnace overhauled, this would require about eight men. In the water-tube plant four men could do the boiler washing very nicely.

There would also be the item of such repairs to the plant that the boiler washers could not do, keeping valves tight, grinding-in check-valves, maintenance of feed and blow-off lines and fittings, but these would not vary greatly with either type of boiler.

To recapitulate, my plan would be to install two rows of eight 500-horse-power water-tube boilers each, with chain grates, and a coal-conveyer system something like that shown by the accompanying sketch. As I figure it, the relative operating expenses of the two plants would be about as follows:

WATER-TUBE PLANT	
2 water tenders @ $75 per mo.....	$150
8 firemen @ $60 per mo.....	480
2 boiler washers @ $65 per mo.....	130
2 boiler washers @ $55 per mo.....	110
1 ash man @ $55 per mo.....	55
5 coal men @ $55 per mo.....	275
Total	$1,200
Total cost for one year.....	$14,400

RETURN-TUBULAR PLANT	
4 water tenders @ $75 per mo.....	$300
32 firemen @ $60 per mo.....	$1,920
2 boiler washers @ $65 per mo.....	130
6 boiler washers @ $55 per mo.....	330
8 ash men @ $55 per mo.....	440
10 coal men @ $55 per mo.....	550
Total	$3,670
Grand total cost for one year..	$44,040
Difference in cost	$29,640

Assuming a life of fifteen years, the difference would be $444,600, which is worth considering.

In the above estimate I have stuck to Mr. Diman's figures, I believe, except in regard to the ash men, and I do not think I have allowed too many men for the purpose, as I have wheeled a few ashes myself and have also interviewed the colored gentlemen who preside over the ash departments of several Kansas City plants.

A reduction in the cost of belting maintenance for a large railroad-shop plant of from $1000 to $250 a month is reported as an actual performance at the Topeka shops of the Atchison, Topeka & Santa Fe road.

Water-tube vs. Horizontal Tubular Boilers

In a recent talk to the Illinois Clay Workers' Association, M. T. Goss, president of the Goss Engineering Company, of Chicago, called the comparative merits of the horizontal-tubular boiler into question, as follows:

How many of you are now running your engine at less pressure than you should have for economy, owing to defective sheets? How many of you have been told by the authorities to put in new boilers because of this? If you buy new tubular boilers will you charge the entire cost to the maintenance account? If not, you should, because had you bought water-tube boilers in the first instance, made for a higher pressure, you could have used them for your required pressure longer than both old and new plants of horizontal-tubular boilers will be allowed to be used for the same pressure, and when this is considered there will be a large credit to the maintenance account of water-tube boilers.

The impression is given by some that water-tube boilers cost from 25 to 40 per cent. more than horizontal-tubular boilers. This is approximately correct for small units, but it is a fact that a 200-horse-power water-tube boiler made for 150 pounds pressure can be made ready for steam for no greater sum than is required for the 100-horse-power horizontal-tubular boiler made for 125 pounds pressure, and for 500 horse-power or more the cost is less.

I have now reached the proposition that a boiler must be so designed that a thick fire-brick arched dutch-oven furnace can be built under the boiler, not over 7 feet wide in any case, and sufficiently long to admit of a thorough mixture of the gases from the volatile matter with the oxygen from the inflowing air, within a temperature sufficiently high for ignition, thus attaining good combustion, and an economical result, as well as a smokeless condition.

It must, however, be borne in mind that to secure these benefits and produce such proper mixture of volatile gases and oxygen it is necessary not to overtax the capacity of the furnace and combustion chamber. It is seldom that more than 50 per cent. efficiency is maintained with hand-fired horizontal-tubular boilers, or with water-tube boilers when the flame passes directly from the fire against the heating surface, because of incomplete combustion due to chilling the gases below the ignition point before the gases from the volatile matter become thoroughly mixed with the oxygen. It is seldom that less than 65 per cent. is maintained with hand-fired water-tube boilers having a long dutch-oven furnace under them. Seventy per cent. is often maintained, and at times 75 and 80 per cent. are attained.

The difference between 50 per cent. and 65 per cent. efficiency represents a loss of exactly 30 per cent. of fuel. An approximation of the yearly loss on this basis in the United States due to incomplete combustion alone, amounts to more than fifty million dollars.

If these statements are true and they can be proved, a water-tube boiler and furnace such as I advocate would be cheaper at a reasonable price than would horizontal-tubular or water-tube with bare tubes over the fire at a very much less price; but, as stated previously, they cost but little more than horizontal-tubular for small units, no more for 200 horse-power, and less for larger plants.

Experience has proven the impracticability of placing long fire-brick surfaces under horizontal-tubular boilers, or any other fire-tube boiler, because if sufficiently long to attain good combustion the heating surface so covered is lost and the extremely high temperature entering the tubes at the rear head will and does invariably sooner or later cause leakage of tubes, which have to be replaced after being rolled a few times.

If fire-tile is placed on the lower row of tubes of water-tube boilers of sectional-header or box-header type, good combustion can be received; the heating surface so covered is lost, the circulation of the hot gases around the lower tubes at the forward end, is not good. If heavy enough to be durable the tendency is to bend the tubes and the expense of removal and replacing when tubes are replaced is considerable. All of the above objections are entirely overcome by the use of a fire-brick arched dutch-oven furnace under a water-tube boiler designed for its installation.

National Society for the Promotion of Industrial Education

This society, which was organized last November, has for its object the bringing to public attention the importance of industrial education as a factor in the industrial and educational development of the United States, by providing opportunities for the study and discussion of the various phases of the problem, and the establishment of institutions for industrial training. The board of managers of the society comprises prominent men of affairs, employers, representatives of labor, social students and educators. The society will depend upon contributions from members, and anybody may become a member by addressing the secretary, C. R. Richards, Columbia University, New York.

Iron sheets coated with aluminum are now being manufactured in considerable quantities and have been found to be very durable under long exposure.

Gas Analysis with Home-made Pipettes

Directions for Making the Apparatus, the Materials Required for the Purpose, and How to Prepare It for Use

BY HARRY SPURRIER

The recognition, by a rapidly increasing number of engineers, of the importance of possessing knowledge regarding the composition of boiler-flue gases has, doubtless, made many of them long for the expensive Orsat apparatus. Unfortunately, most proprietors of steam plants object to the cost of scientific appurtenances of this character, little realizing that every pound of carbon burned to carbon monoxide instead of to carbon dioxide means a loss of nearly 70 per cent. of the heat of each pound of carbon, and that analysis of the

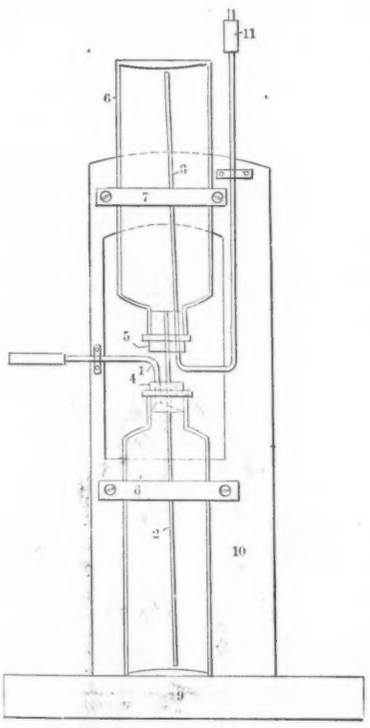


FIG. 1. CONSTRUCTION OF SIMPLE PIPETTE

flue gases is the only means of detecting this, as well as other furnace losses. Finding it absolutely necessary to make analyses of flue gases, and not being able to secure either the Orsat device or the "regulation" gas pipettes, I determined to devise a substitute, and accordingly rigged up the pipettes which are here described.

These pipettes are not "some sort of a makeshift for a rough guess," but if carefully made and used with care they will work accurately. I shall only go into the subject of flue-gas analysis so far as is necessary to make the construction and use of the pipettes intelligible, especially since gas analysis has lately been ably expounded to POWER readers by Professor Gill. To make my apparatus one simple and two composite pipettes are required, in addition to a gas burette and a leveling tube, or bottle. The burette will cost about three dollars and will have to be purchased. The leveling bottle arrangement can easily be made at home.

To make the pipettes the following materials will be necessary: Ten four-ounce clear glass oil-sample bottles, ten stoppers each with two longitudinal perforations, one-half pound of quill glass tubing, three feet of capillary glass tube, two yards of good rubber tube to fit the glass tubes snugly, three pinch-cocks, six wood screws, eight stove bolts, a small glass funnel and a few pieces of wood.

THE SIMPLE PIPETTE

For the simple pipette, Fig. 1, use two oil-sample bottles and two of the rubber stoppers, which may be cut down if too long. It is essential that the stoppers fit tightly, as shown in *B*, Fig. 2. The stopper shown in *A*, Fig. 2, is cut the wrong way. With a keen thin blade cut out a cone from the small end of one of the stoppers, the apex of the cone occurring at the opening of one of the vertical holes (see Fig. 2). Take a piece of the capillary glass tube *1*, Fig. 1, and make a right-angle bend about one inch from the end by turning the tube round and round in a rather small bat's-wing burner flame, until it shows a decided tendency to bend over by its own weight; then stop turning and allow it to bend downward until the two legs are at right angles; do not attempt to force it, and do not heat up too quickly. Leave the soot on until the tube is quite cold, then wipe the tube and cut off all but about one and one-half inches of the straight section below the bend. To cut the tube, wet one edge of a fine triangular file with turpentine and with it make nicks in the tube at opposite points. Grasping one end of the tube in each hand place a thumbnail in one nick and press upon it, at the same time drawing asunder with the hands; a clean break will result. Trim off the sharp edge with the flat of the file, which has first been moistened with turpentine. Insert the shorter leg of the tube in the rubber stopper *4*, Fig. 1, with the end of the tube flush with the apex of the cone.

Pass a quill tube *2*, Fig. 1, through the other hole of the stopper, and putting the stopper in the bottle, push the quill tube down to within 1/16 inch of the bottom. If the quill tube and the wall of the hole are moistened with alcohol or soda solution, the tube will slip through the stopper more easily. Insert another stopper *5*, Fig. 1, large end first into an inverted bottle, extend the upper end of quill tube *2* into the corresponding hole in this stopper, and cut the tube off flush with the small end of the stopper. Take another length of quill tubing *3*, Fig. 1, and at a sufficient distance from one end make a bend by means of the bat's-wing burner, supporting the tube with one hand on each side of the flame (the tube should be held parallel with the flame), and turning in the flame until the tube softens allow it to bend over until the two legs are at right angles. Allow it to cool off a little, then heat up again and turning the tube round and round form a second bend about one inch from the first, having the legs of the tube parallel so that it forms a U-tube. When cold pass the short side of the U-tube through the stopper and cut off the outer leg about one inch above the bottom of the inverted bottle.

The stand *9*, Fig. 1, for this pipette needs

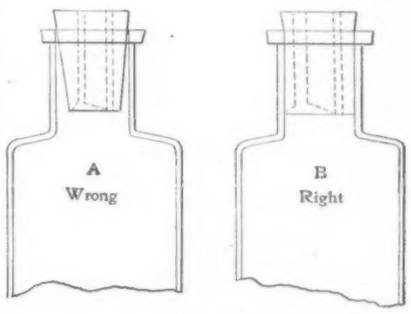


FIG. 2

but slight description. The base is best made of material three inches square through section. Split along the grain into two pieces, one 2x3 inches, the other 1x3 inches. A piece of 1¼-inch board *10* of sufficient width, with a panel cut out as shown, is set up against the 2x3-inch pi ce, the 1x3-inch piece placed behind it, and a couple of wood screws driven in from the back so as to hold all three pieces firmly together, thus forming the supporting frame for the apparatus. The opening in the frame facilitates observation of the action going on in the bottles and tubes. In Fig. 1, *7* and *8* are pieces roughed out on the reverse side to fit

the curve of the bottles and are held at each end by a stove bolt. A short piece of rubber tube *11* plugged at one end with a projecting section of glass tube (this short tube being sealed up by turning it round and round in a Bunsen flame until it closes entirely) is slipped over the tube *3*, Fig. 1. Small cleats of cork hold the

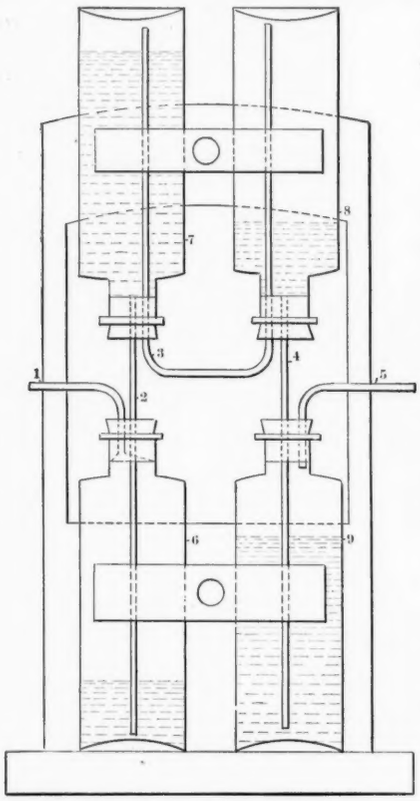


FIG. 3. DIAGRAM OF THE COMPOSITE PIPETTE

tubes *1* and *3* fast to the support, a wedge being placed behind the tubes to keep them parallel with the back of the support. The completed pipette is now rigid and can be shaken without danger.

The Composite Pipettes

This style of pipette, two of which are needed. is shown in Fig. 4. The lower left-hand bottle is for the reagent and absorption of the gas, while the bottle immediately above it is to hold the absorbing reagent as the entering gas drives it out of the lower bottle. The upper right-hand bottle is the air-trap and the lower right-hand bottle serves as a water-seal to prevent the entrance of air. This pipette has (with three others) been in use for more than two years and has "filled the bill" so completely that I have never felt the need of an Orsat apparatus.

Proceed to make this pipette in exactly the same way as the simple pipette. Referring to Fig. 3, it will be seen that the U-tube *3* is a little wider at the base, but the legs are of the same length as the tube in Fig. 1. After inserting one leg of this tube in the bottle *7*, Fig. 3, slip another stopper, large end first, over the free leg and place the bottle *8* firmly in position. Take another stopper and pass a piece of quill tube *4* through it to the bottom of bottle *9*, then putting this bottle in place determine the point at which to cut the tube *4*. With this tube in place form quill tube *5*, and insert it in the stopper as shown. Keep the work lying flat on the table while doing this.

The frame support for this pipette is easily and quickly made; one bolt is sufficient to hold each of the wooden cleats and if the bolt-holes are a little large the external pressure upon the bottles will be evenly divided.

To test the pipettes place a few drops of water on the rubber stopper and closing either tube with a rubber cap or pinchcock, blow into the open end; if air bubbles make their appearance there is a leak which must be stopped, as absolute tightness is essential. The stoppers should all be carefully tested in turn. If difficulty is experienced in securing perfect freedom from leakage the stoppers may be covered with paraffin wax, softened with oil to prevent the wax becoming brittle when cold. Put the wax on in small pieces and spread it around the joints with the heated tang of a file.

To Fill the Pipettes

When all joints have been proved to be quite tight proceed to fill the pipettes as follows:

The simple pipette is filled with a solution of caustic soda; a good quality of commercial caustic soda put up in air-tight cans will answer. Take two ounces of dry caustic (handle carefully), dissolve it in about nine ounces of water in a bottle, cork the bottle and allow it to settle thoroughly. Attach a piece of rubber tube not less than one foot long to the end of tube *1*, Fig. 1, and attach the other end to the tail of the glass filling funnel. By holding the funnel up the solution can be poured directly into the pipette, being careful to avoid the introduction of sediment. Pour in enough solution to fill the lower bottle and to just cover the shoulder of the upper one. Now put a cap at *11* and also one on *1*. The pipette is now ready for use; label it "No. 1 for Carbon Dioxide."

To fill the composite pipettes, dissolve one ounce of pyrogallic acid in two ounces of water and pour this, by aid of the previously washed funnel, into bottle *6* of the composite pipette, Fig. 3. Through the same funnel next pour in enough caustic soda solution to fill the bottle and also cover the shoulder of the upper bottle *7*, Fig. 3. Now by sucking at *5* (use a piece of rubber tube), draw the solution out of the lower bottle into *7* and put a cap (rubber tube and plug) on *1*. Connect the washed funnel and its tube on *5* and fill with water; take the cap off of *1* and the water will run into bottle *9*, up tube *4* and into bottle *8*. At the same time the solution in bottle *7* will run down into bottle *6*. When bottle *6* and tube *1* are completely filled, put caps on *1* and *5* and label this pipette "No. 2 for Oxygen."

The composite pipette for carbon monoxide is filled with a saturated solution of cuprous chloride (not cupric chloride) in hydrochloric (muriatic) acid. Take rather more than enough strong hydrochloric acid to half fill one of the bottles, dilute with its own volume of water, add cuprous chloride as long as any will dissolve, and when a further addition of the chloride fails to dissolve on shaking, allow the solution to settle (this mixture should be made in a glass-stoppered bottle). The solution can be fed into the pipette in the same way as the previous one, making the water-seal and closing *1* and *5* by small rubber caps. The liquids used in these pipettes are corrosive and dangerous and must be handled with caution.

A very handy apparatus to employ in securing a sample of gas is made up of an ordinary cone-topped two-quart can provided with a doubly perforated stopper and two glass tubes bent at right angles, the short side of one tube being flush with the small end of the stopper, the other tube extending to within one-quarter of an inch of the bottom of the can. Fill the can and tubes with water, connect the long

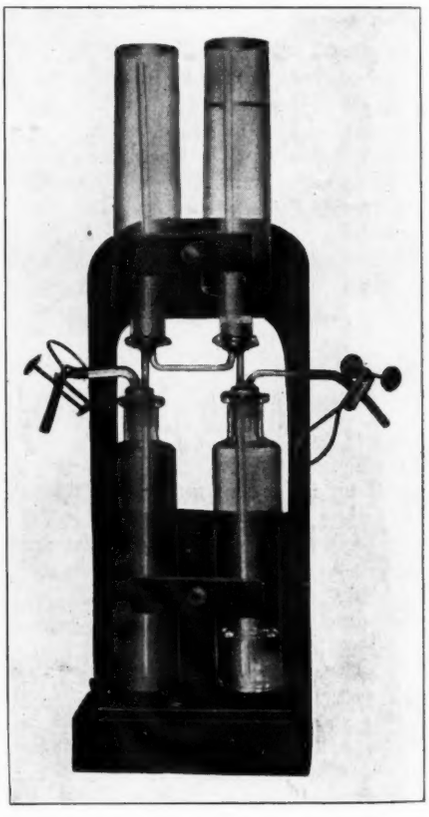


FIG. 4. THE COMPOSITE PIPETTE

tube with the source from which the gas is to be drawn, attach a piece of rubber tubing on the shorter glass tube, invert the can and start the water running with a gentle suction on the rubber tube. Allow the water to run out until the can is nearly but not quite empty, then close the two tubes and disconnect. The sample

may then be taken home and kept until required for analysis.

In order to transfer a sample of gas to the burette, shown in Fig. 5, first connect the upper tube with the short tube on the can. Fill the burette and rubber tubes completely full of water and close the pinch-cock on the upper tube. Insert in this tube a short piece of glass tube also filled with water. With one hand compress the rubber tube on the short bend in the can quite close to the pinch-cock; slide the fingers along the tube (moistened with a drop of weak soda solution, if necessary) to the end; now insert the short glass tube in the compressed rubber tube in such manner as to exclude every bubble of air. This done, open the pinch-cock (by slipping it over the glass tube) between the level bottle and burette. Next insert the tail of the funnel in the tube connected to the longer tube in the can; fill the funnel with water and thrust a small stick into the tail of the funnel to dislodge the air. Allow the water to run into the can and this will drive the gas into the burette. Let the flow of gas into the burette continue until it is emptied of water; do not allow the funnel to run empty, or air will get into the can. Close the upper tube by means of the pinch-cock, raise the level bottle until the water re-enters the burette and is just level with the 100 c.c. mark on the burette. Now the burette is full of gas and under a slight pressure. Open the upper pinch-cock for an instant to equalize the pressure in the burette with the atmosphere, when the burette will be full of gas at atmospheric pressure.

Introducing the Gas into the Pipettes

To introduce the gas into the simple pipette, moisten the forefinger and thumb of the left hand, and taking hold of the upper connecting tube of the burette close to the pinch-cock, slide the fingers toward the end of the tube, excluding the air, and slip it over the capillary tube of the pipette; the soda solution in the pipette having been brought right up to the tip of the capillary tube and kept there by capping the tube *11*, Fig. 1. Now open and slip the pinch-cock on the rubber connection between the burette and the pipette, over the glass tube, thus keeping the pinch-cock open. Take the cap off of *11*, Fig. 1, raise the leveling bottle until the water which then flows into the burette completely fills it and drives the gas into the pipette, then close the pinch-cock on the rubber tube again, and take the rubber tube off of the burette. Agitate the pipette for about five minutes to absorb the carbon dioxide, then remake the connection between the burette and pipette, taking the precaution to exclude all the air. This done, open the pinch-cock and by lowering the level bottle the water level in the bottle will sink, drawing in the unabsorbed gas. This drawing-in should continue until the last bubble of gas has left the pipette and the soda solution just shows in the top capillary of the burette. Close the pinch-cock on the top rubber tube and bring the level bottle up until the level of the water in the bottle is exactly even with the level of the water in the burette. Holding them closely beside each other, read the volume. The difference between this reading and one hundred equals the percentage of carbon dioxide absorbed.

In a precisely similar way the two composite pipettes are manipulated, first passing the gas into "No. 2 for Oxygen," shaking well for about five minutes to allow the absorption to become complete, and then returning to the burette for measurement. The same process is employed in connection with pipette "No. 3," for the absorption of carbon monoxide. The unabsorbed residue is counted as nitrogen. To successfully connect and disconnect the tubes requires a little practice to avoid the introduction of air or the escape of gas, but one or two trials with air alone is quite enough to establish sufficient dexterity. The working of the glass seems to offer prohibitive difficulties to some persons, but a little practice (and above all one or two failures) will teach more than can be easily written. The knack in these things is soon acquired and yields abundant pleasure.

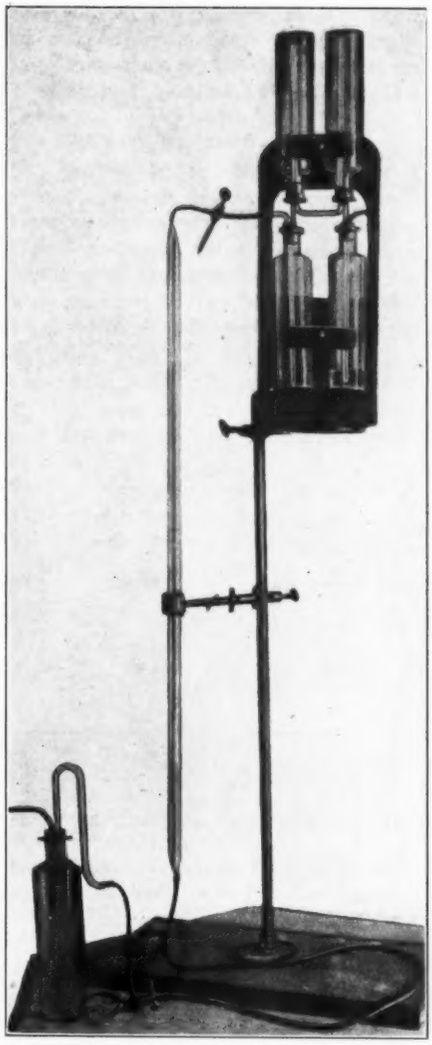


FIG. 5. COMPLETE OUTFIT, WITH BURETTE AND LEVEL BOTTLE

To reduce friction in working rubber always use a little alcohol, and in filing glass use turpentine, as it saves the file and works miracles in cutting tubes, etc. The burettes sold by chemical-glass dealers are usually accompanied by the leveling tube and stand, which is quite convenient. The burette should be protected carefully from any change in temperature while in use, as the ratio of expansion for gases is 1/273 of their volume for each rise in temperature of 1 degree Centigrade.

Lap-joint Boilers

In commenting upon the Lynn explosion, and also upon the previous explosion at Brockton, a number of persons, including some well known engineers, have suggested that the use of the lap seam be absolutely prohibited by law, the butt joint, which apparently is not liable to similar cracks, being required in its place. This appears to us to be a rather drastic proposition. We should like to see the lap seam discarded altogether, for boilers carrying considerable pressures; but we are aware, at the same time, that the lap joint is regarded by many as entirely satisfactory, when it is well designed and well constructed. At all events, designers of admitted ability continue to make use of it, as they would not do if they were satisfied that it was markedly dangerous. It is probable that eighty per cent. of all the boilers now in use in the United States have lap joints; and the proposition to make the further use of these boilers illegal strikes us as calling for the most serious consideration before acceptance. Certainly there never has been (so far as we are aware) any municipality or State or government, nor any bureau of steam engineering, which has yet made any permanent, wholesale and unqualified condemnation of the lap joint. In view of these facts, it is certainly reasonable to suggest that the matter be most carefully considered before manufacturers and boiler owners generally are put to the enormous expense and inconvenience that would be involved in prohibiting, at once, the use of lap-joint boilers. We are not here pleading for the continued use of the lap joint, but only for reasonableness in any legal enactments that may be contemplated by the legislators of the country.—*The Locomotive.*

The builders' overhaul of the Cunard turbine liner "Carmania" at Liverpool, after her twelve months' service, has been completed, and it is said that the results of the investigation and overhaul proved most satisfactory.

Elementary Electrical Engineering—XV

Effect of Armature Reaction in a Motor, and the Relation between Field Strength and Armature Speed

BY CECIL P. POOLE

From Figs. 38 and 41 in the last two articles of this series the student is apt to get the impression that there are two distinct magnetic fluxes through the armature and field-magnet cores of a motor. This is not true; there is only one magnetic flux, but it is the result of two distinct magnetomotive forces: that of the field magnet and that of the armature winding. As explained in an earlier article, magnetomotive force is the force which tends to produce a magnetic field or flux, just as electromotive force tends to produce an electric current. The magnetomotive force of the field magnet tends to produce a magnetic flux straight through the armature core, as indicated by the line marked "Field M.M.F." in Fig. 42, and if there were no current in the armature winding, the axis of the field flux would follow that line; if the direction of the exciting current through the field winding be such as to produce a north pole at the left and a south pole at the right, as indicated by the letters *N* and *S*, the direction of the field magnetomotive force will be from left to right, as indicated by the arrow head and tail at the ends of the m.m.f. line. Now, if there be no current in the field winding and a current be passed through the armature winding, it will set up a magnetomotive force at right angles with the force that was exerted by the field winding, supposing the brushes to be set on the theoretical neutral line. If the direction of the armature rotation, when the machine is running, be that indicated by the arrow *R*, then the magnetomotive force set up by the armature winding when the field is not magnetized will have the direction indicated by the vertical arrow marked "Armature M.M.F." in the diagram, and this force will produce a flux which will follow the straight line from edge to edge of the armature core and curve over to the field-magnet poles somewhat as indicated by the dotted lines.

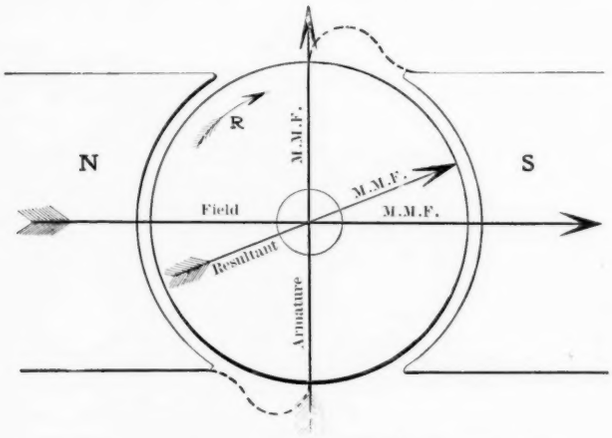


FIG. 42. COMBINATION OF M. M. FS. IN ARMATURE CORE

Effect of Armature Reaction

Now if current be allowed to pass through both the field and the armature windings, the two magnetomotive forces will combine, between the pole-faces of the magnet, into a resultant force the direction of which will be somewhere between the directions of the original two forces; the field magnetomotive force is usually much stronger than that of the armature winding, so that the resultant magnetomotive force is much nearer the direction of the field force than that of the armature; this is indicated by the arrow marked "Resultant M.M.F." However, even if the two primary forces were equal, so that the resultant force would be halfway between them, the axis of the magnetic flux produced by the resultant force could not coincide with the direction of the resultant force, because very little shifting of the flux in the magnet pole suffices to raise the magnetic density in the crowded pole corners to such a high degree of saturation that it cannot be shifted farther by the magnetomotive force available. If there were no iron in the magnetic circuit, the axis of the path followed by the flux would agree nearly with the direction of the resultant magnetomotive force produced by the two windings. In actual practice, the path followed by the flux through the pole-faces and armature is somewhat like that indicated by the long arrow in Fig. 43.

The distortion of the course of the field flux from a straight path across the armature has the same effect as would twisting the whole field magnet around the axis of the shaft, with the flux following a straight path from pole to pole. This necessitates shifting the brushes backward against the direction of armature rotation, and this, in turn, shifts the field flux still more, because it changes the direction of the armature magnetomotive force. A point is soon reached, however, where the saturation of the magnet-pole corners stops the shifting of the flux away from the straight path which it would follow if undisturbed, and this point of stability, in a well designed motor, occurs not very far from the theoretical neutral axis of the armature winding; that is, the actual neutral axis of the armature winding, in operation, is very little behind the neutral axis that it would have if there were no field distortion. This is illustrated in Fig. 43, where the theoretical neutral axis is represented by the vertical solid line and the actual or working neutral axis by the dotted line.

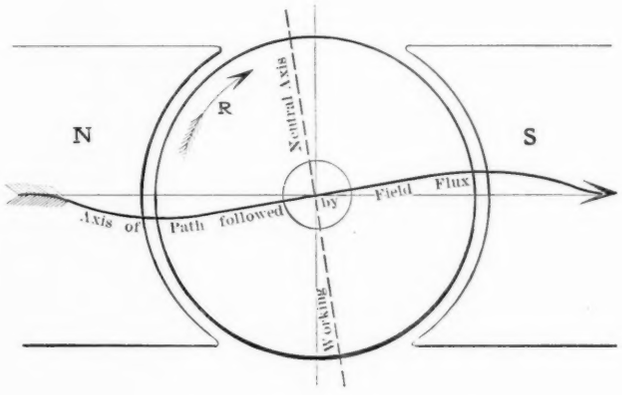


FIG. 43. FIELD DISTORTION BY ARMATURE REACTION

Effect of Field Variation on Speed

As stated briefly last month, weakening the field of a direct-current constant-potential motor will cause the speed to increase almost in inverse proportion up to a certain point beyond which any further reduction in field strength will cause a decrease instead of an increase in the speed. It might seem that no such point could be reached because the impressed e.m.f. is constant and therefore reducing the

field strength must cause the armature to speed up, under all conditions, in order to keep its counter e.m.f. up. It must be remembered, however, that when the field strength is reduced, the armature current must increase so that the product of field flux and armature current will remain the same, if the torque is constant, the torque formula being:

$$\frac{\Phi \times p \times w \times I}{852{,}200{,}000 \times m} = T,$$

in which

Φ = Magnetic lines of force under each pole-face,
p = Number of field-magnet poles,
w = Total number of armature wires,
I = Armature current,
m = Number of paths in armature winding,
T = Pound-feet of torque.

Transposing this formula to read:

$$\frac{852{,}200{,}000 \times m \times T}{\Phi \times p \times w} = I,$$

shows clearly that with a constant torque any decrease in the field flux Φ will result in a proportionate increase in the armature current I. Thus, if the field strength be reduced to one-half of its normal value, the armature counter e.m.f. will be momentarily reduced to one-half and the current will be momentarily increased to more than twice its former strength.

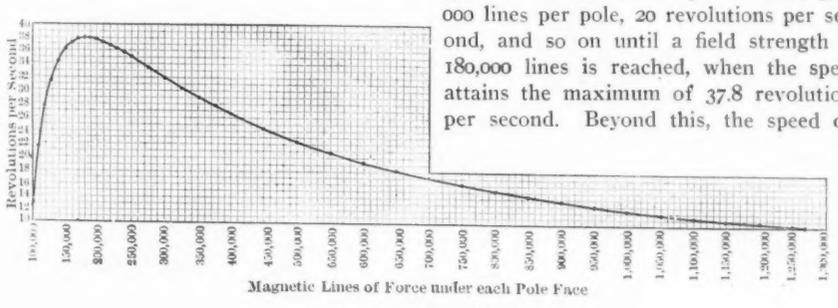


FIG. 44. A FIELD-TORQUE CURVE

The speed will immediately increase to a point where the counter e.m.f. is such as to allow twice the original current to pass through. For example, suppose that when running normally the drop in the armature winding is 5 volts and the impressed e.m.f. is 110 volts. The counter e.m.f. will be 105 volts. Now let the field strength be reduced to one-half its original value; the counter e.m.f. will drop momentarily (assuming the speed not to have changed) to 52.5 volts; the armature current required to keep up the torque will be doubled, so that the drop will be increased to 10 volts, leaving 100 volts to be balanced by the counter e.m.f., and the speed will immediately increase to that required to generate 100 volts. With the original field strength, the speed necessary to generate 100 volts would have been 100/105 or 0.9524 of the original speed; consequently, with the field strength reduced one-half, the speed necessary to generate 100 volts will be $0.9524 \times 2 = 1.9048$ times the original speed. Further reduction of the field strength will cause the speed to increase in smaller and smaller proportion until a condition of equilibrium is reached, when any further decrease in field strength will cause a decrease, instead of an increase, in speed. This is made clearer by reference to a diagram showing the relation between the field excitation and the speed, the torque remaining constant.

Such a diagram is given in Fig. 44; this is the excitation-speed curve of a 7½-horse-power 110-volt motor running at 6.7 horse-power with normal field excitation and corresponding outputs with other excitations. The scale along the bottom of the diagram gives the number of magnetic lines of force under each pole-face and the scale at the left the number of revolutions per second. Tracing up from any point on the bottom scale to the curve and thence horizontally to the side scale will give the speed at which the motor will run when the field strength is that indicated at the starting point on the bottom scale. Thus, with 1,000,000 lines per pole, the speed is 12½ revolutions per second; with 1,140,000 lines per pole, the speed is 11 revolutions per second, and with 1,270,000 lines, the speed is 10 revolutions per second. Going the other way, it will be found that a field strength of 760,000 lines per pole gives a speed of 16 revolutions per second; 575,000 lines per pole, 20 revolutions per second, and so on until a field strength of 180,000 lines is reached, when the speed attains the maximum of 37.8 revolutions per second. Beyond this, the speed decreases rapidly with decreasing field strength.

Plotting a Field-speed Curve

It will be of great assistance to the student to plot excitation-speed curves of several imaginary motors, taking as widely different conditions as possible, especially as regards armature resistance. It will be found that the higher the resistance of an armature, other conditions being unchanged, the less can the field strength be reduced before reaching the critical condition indicated by the crest of the curve. The formula for plotting the curve is as follows:

$$\frac{j}{\Phi} - \frac{k}{\Phi^2} = s,$$

in which

Φ = Number of magnetic lines of force under each pole-face.

$$j = \frac{100{,}000{,}000 \times E\,m}{w\,p}.$$

$$k = \frac{85{,}220{,}000{,}000{,}000{,}000 \times T\,m^2\,r}{w^2\,p^2}.$$

s = Revolutions per second.

The meanings of the symbol letters in the expressions for the values of j and k are as follows:

E = Impressed e.m.f. at motor terminals,
m = Number of paths through armature winding,
w = Number of armature wires around core periphery,
p = Number of field-magnet poles,
T = Torque (pound-feet),
r = Resistance of the armature circuit, including brushes and their connections.

All of these factors may be ascertained from an actual motor without making any test, excepting the armature resistance, and that can be easily determined by passing a small current through the armature while it is held stationary and measuring the volts at the brush connections; dividing the volts by the amperes of current will give the resistance. The torque may be calculated for normal conditions by means of the formula:

$$87.535 \times H.P. \div s = T,$$

in which $H.P.$ represents the horse-power delivered at the pulley of the motor and s the revolutions per second when delivering that horse-power. It is advisable to assume the motor to be working at full load and rated speed when the field strength is normal; then calculate the torque and measure the armature resistance.

Example

Suppose a 240-volt motor to be rated at 930 revolutions per minute at full load and to deliver 10 horse-power at that speed. The revolutions per second will be 15½, and the torque will be:

$$87.535 \times 10 \div 15.5 = 56.474 \text{ pound-feet.}$$

Ordinarily there would be no need to carry an engineering calculation to three places of decimals, two being ample for most cases; here, however, it is advisable to use the three places because it is desired to compute values in which small deviations from accuracy may invalidate the result.

Now suppose that when a current of ten amperes is passed through the armature, with the latter held stationary, the voltage at the brush-holders is 6.49; then the resistance of the armature circuit will be 0.649 ohm. If the number of armature wires is 690, the number of field-magnet poles 4, and the number of paths through the armature winding 2, the value of j will be

$$\frac{100{,}000{,}000 \times 240 \times 2}{690 \times 4} = 173{,}913{,}000$$

and the value of k will be

$$\frac{85,220,000,000,000,000 \times 56.474 \times 4 \times 0.649}{476,100 \times 16} = 1,640,123,700,000.$$

Now, by assuming different values for the field flux Φ, beginning say with 100,000, and taking values 10,000 greater each time, the speed corresponding to each value of Φ can be computed by the formula

$$\frac{173,913,000}{\Phi^3} - \frac{1,640,123,700,000}{\Phi^3} = s.$$

Take a piece of cross-section paper and lay out the scale of field strengths along the bottom, and the vertical scale of speeds along the side in a manner similar to that illustrated by the chart, Fig. 44. Then at intersections corresponding to the related values of Φ and s make small dots. When the curve begins to flatten out toward the right-hand end, the values of Φ may be taken farther apart, say every 50,000 lines instead of every 10,000. When all the dots are in place, draw the curve through them. The dots which determined the curve in Fig. 44 are shown in the diagram. The data used for plotting the diagram were as follows:

Impressed voltage	110
Torque (constant)	46.94
Speed at that torque	12.5
Horse-power at that speed and torque	6.7
Total number of armature wires	400
Number of field-magnet poles.	4
Number of paths through armature winding	2
Resistance of armature circuit	0.2

After plotting a curve from the data of an actual motor, it will be instructive to assume a larger armature resistance, keep all other data unchanged, and plot another curve.

The reader must not be misled by the foregoing discussion into the belief that a motor can be operated at any desired speed by adjusting the field strength. In practice, other considerations come in which limit the extent to which the speed can be controlled by varying the field strength. The principal obstacle to unlimited speed control by field variation is sparking at the brushes, which occurs when the field is weakened beyond a certain point relative to the "magnetic strength" of the armature. This will be discussed fully in a future article.

Air Compressors on Man-o'-War Reef

By Frank Richards

The half-tone, Fig. 1, shows an interesting air-compressor installation—that of the Degnon Contracting Company—on what is known as "Man-o'-War Reef" in the middle of the East river opposite Forty-second street, New York. The existence of the reef in line with the proposed Belmont tunnels suggested the practicability of sinking shafts there and driving the tunnels from them in both directions, thus doubling the number of working faces and presumably halving the time required for the completion of the work.

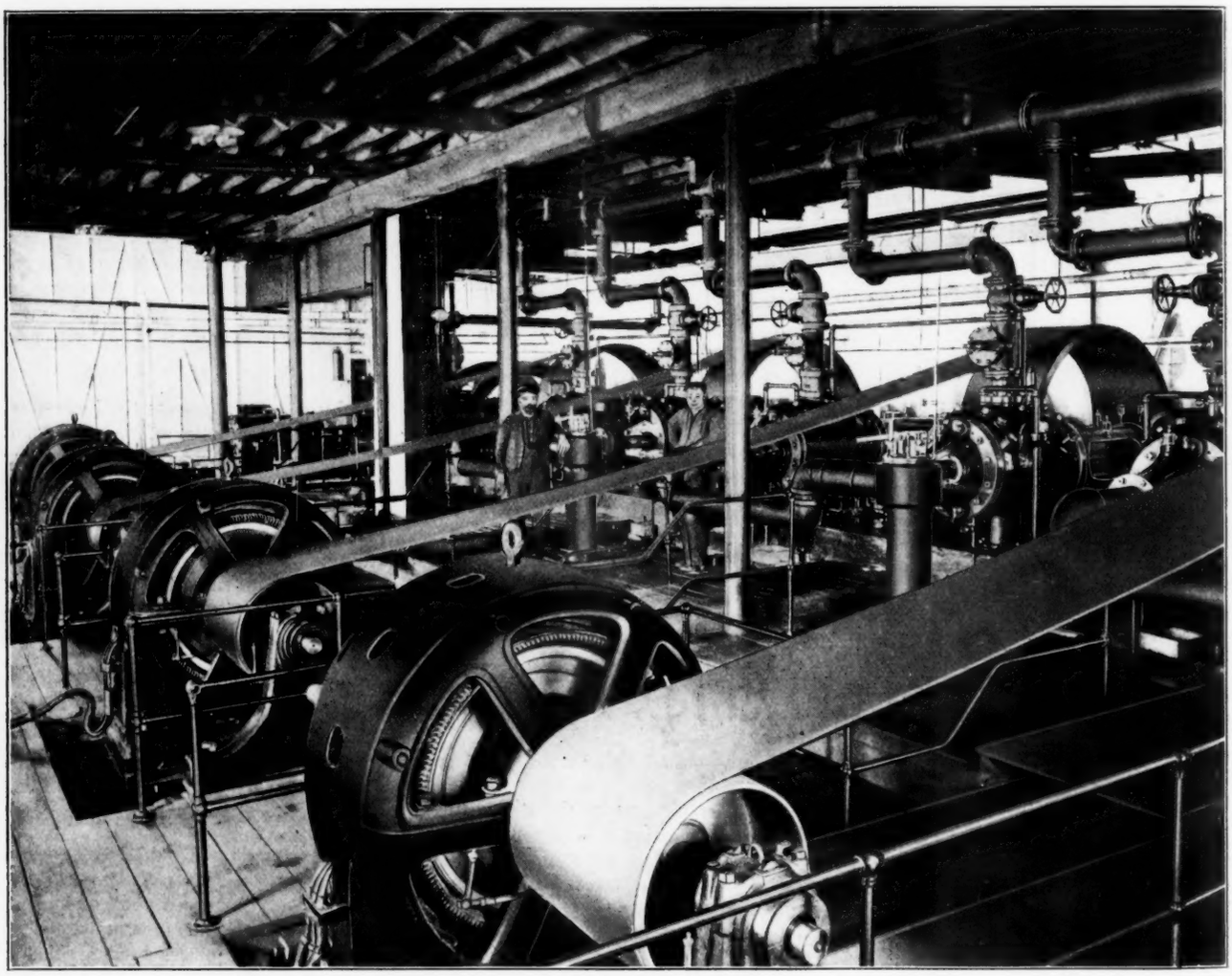


FIG. 1. INTERIOR OF AIR-COMPRESSOR HOUSE ON MAN-O'-WAR REEF, EAST RIVER, NEW YORK

Originally but a small area of the reef showed above the surface of the water, and the first thing was to enlarge it; or, in fact, to build an island of sufficient size to erect a plant upon. A single boiler and a straight-line air compressor were used to drive rock drills for the preliminary work of shaft sinking, and the first material taken from the shafts was used for filling upon and around the reef. When

the area was sufficiently enlarged the compressors were installed as shown. The compressor house extends close to the docked edge of the little island on the east and the south. On the New York side there is a little more than standing-room outside the building, while on the north end of the island (Fig. 2) sufficient land was made to accommodate tracks for the small narrow-gage cars which carry the excavated material to the scows.

Referring to Fig. 1, the most distant compressor is a straight-line machine, 24 inches diameter steam and 26¼ inches diameter air, by 30 inches stroke; free air capacity, 1843 cubic feet per minute. There is another compressor similar to this at the right and somewhat to the rear of the spectator, so that it is not included in the view. Between these two are the four belted electric-driven compressors as shown. Three of these are duplex machines with air cylinders 20¼ inches diameter by 16 inches stroke; aggregate free air capacity, 6540 cubic feet per minute. The fourth of the electric-driven machines is a two-stage compressor with air cylinders 25¼ inches, and 16¼ inches diameter by 18 inches stroke; free air capacity, 1704 cubic feet per minute. This last named compressor delivers the air somewhat above 100 pounds, while the maximum for the other three is 50 pounds. These are Ingersoll-Sergeant and Ingersoll-Rand machines. They all run at constant speed, the air delivery being regulated by choking controllers on the intake. The three low-pressure machines run quite constantly at full capacity, the work apparently taking all the air available. The boiling up of the water over the tunnel heading, which cannot be fairly realized from the view shown in Fig. 3, indicates where the air goes.

The current for the electric drive is taken from a cable connecting with the lines of the Interborough Rapid Transit Company. The four motors are Westinghouse induction motors, each of 200 horsepower, 2300 volts, three-phase, 25-cycle, running at 480 revolutions per minute. They are arranged with three bearings, base-plate and slide-rod adjustment. The guaranteed efficiency at full or three-quarter load is 89 per cent., and at half load 87½ per cent. They operate at full load rating with a temperature rise not exceeding 40 degrees Centigrade; at an overload of 25 per cent., 50 degrees, and at 50 per cent. overload for one hour, not exceeding 60 degrees.

It will be noticed that chiefly to economize space in placing these motors, the belts are led backward over the air-cylinder ends of the compressors instead of in the other direction, as has been the custom. The present arrangement is more compact and at the same time gives belts of good length. The belts run very straight and smooth. Electric motors are used also for the hoists in the two shafts. Two boilers of locomotive type supply steam for the straight-line machines, only one of them usually running. The feed-water for the boilers is piped from Manhattan.

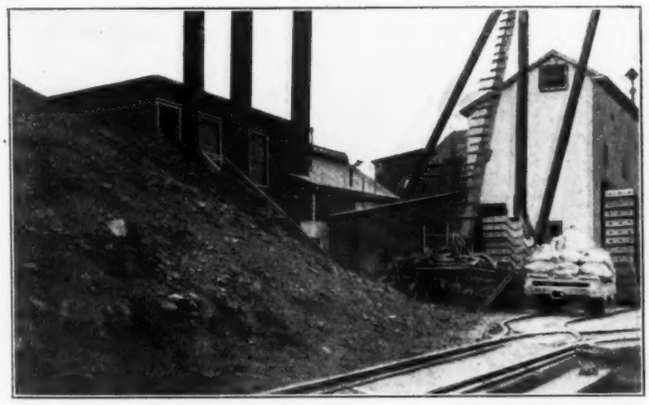


FIG. 2. NORTH END OF MAN-O'-WAR REEF EMBANKMENT

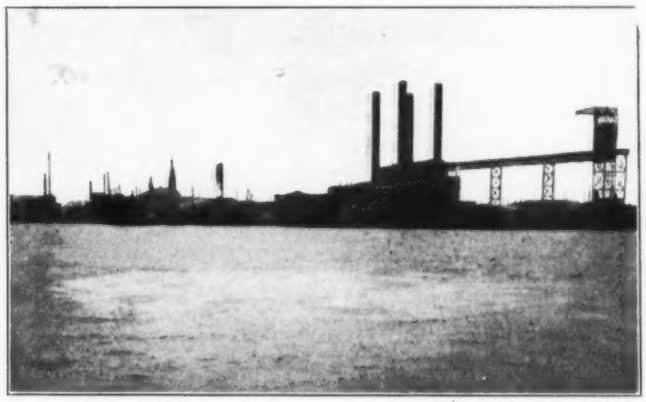


FIG. 3. SHOWING LEAKAGE OF AIR OVER TUNNEL HEADING

It may be easy to assume that this emergent compressor plant, on account of its location and the exacting conditions to be more or less sacrificed to, must necessarily be a costly plant to operate, but this is hardly so. The electric-driven machines in this case, which do most of the work, take current from a service in which the highest possible economies are attained, and having motors adapted to their work deliver the air at a lower cost than that of the direct-steam driven machines beside them, notwithstanding that the latter represent still widely prevalent practice.

For Power Development at Niagara Falls

It was recently announced that Secretary of War William H. Taft has decided, with reference to the generation and transmission of electric power at Niagara falls, that 15,600 cubic feet of water per second may be diverted from the American side and 160,000 electrical horsepower from the Canadian side, without substantially impairing the scenic beauty of the falls. He has already granted four permits for the transmission of 160,000 electrical horse-power in the aggregate, and of this total the International Railway Company, of Buffalo, will be permitted to transmit 1500, the Canadian Niagara Power Company 52,500, the Ontario Power Company 60,000, and the Electrical Development Company 46,000 electrical horse-power. The International Railway Company asked for 8000 horsepower, the Ontario Power Company for 90,000 horse-power, the Electrical Development Company for 62,500 horse-power, and the Canadian Niagara Power Company for 121,000 horse-power.

The Canadian Niagara Power Company will deliver its power to the Niagara Falls Power Company over transmission lines across the Niagara river at Niagara falls and Buffalo.

The Ontario Power Company will deliver power to the Niagara, Lockport and Ontario Power Company, whose transmission line crosses the Niagara gorge below the Whirlpool and extends eastward to Rochester and Syracuse.

The Electrical Development Company will deliver power to the Niagara Falls Electrical Transmission Company, which has yet to build its transmission line across the river; but plans are made for a crossing over a new steel bridge, to be erected close up by the falls.

The Niagara Falls Hydraulic Power and Manufacturing Company is now using 4000 cubic feet of water per second on the New York side, but as it had contracted for the sale of a large block of power to the Pittsburg Reduction Company long before the Burton bill was passed, the company is given a permit to divert 6000 cubic feet of water per second, which will supply its new power station now building.

A committee has been appointed, also, to suggest changes to be made in existing plants on the American side just below the falls, to restore as far as possible the scenic attractiveness of the cañon.

Practical Letters from Practical Men

Don't Bother About the Style, but Write Just What You Think, Know or Want to Know About Your Work, and Help Each Other

WE PAY FOR USEFUL IDEAS

"The Steam Plant of the Wood Worsted Mill"

In the very interesting article and editorial comment on the Wood Worsted Mill power plant it appears that the 20 boilers (4000 horse-power) burn about 100 tons of coal daily.

The cost of horse haulage from the coal pocket to the firing floor is given as $3000 per year—only $10 a day, or 10 cents per ton, involving a charge of 75 cents per horse-power a year. As a tip-cart holds a ton of coal, this means 100 trips a day, involving at least two shifts each of one driver and two horses, and it is difficult to see how this can be done at $10 a day.

But in addition it takes at least two shifts each of eight firemen and four men at the coal pocket to fire and handle the 100 tons of coal daily, a total of 24 men extra; taking their wages at only $2.50 a day, this represents a labor charge of $60 daily, or $18,000 yearly, or $4.50 per horse-power per year.

It is perfectly clear that with a coal conveyer, without an elevated coal pocket, two men could handle the coal to the conveyer as compared with four men loading coal into carts; and also that four men could easily attend to the 20 boilers with mechanical stokers, instead of eight men needed for hand firing.

The installation of a coal conveyer and mechanical stokers should therefore easily save the 75 cents per horse-power for horse haulage, and half the other labor cost, making a total labor saving of, say, $3 per horse-power per year.

The cost for the 20 mechanical stokers and for the coal conveyer to handle 100 tons daily could hardly exceed $20,000, or $5 per horse-power; so that mechanical handling and firing should easily repay the entire investment within two years and give an investment returning about 60 per cent. per annum. The fuel saving due to mechanical firing would at least cover the cost of operating the conveyer and stokers, even with the same coal.

The completed cost of the 8000-horse-power plant is to be $5 per horse-power less than the final estimate; and as, according to the table in your editorial, the average cost of New England power plants involves standing charges of $4.72 per horse-power, which on your basis of 12 per cent. would indicate that the average plant of this type costs only about $40 per horse-power; so, if the Wood plant costs, say, $35 per horse-power, the difference appears to be represented by the absence of the coal conveyer and mechanical stokers, which would save $3 a year in labor on the investment of $5 per horse-power, which is hardly a desirable form of reducing first cost.

It therefore appears, not that the cost of this plant is "ridiculously low," but that the original scheme, involving twice the cost per horse-power, was ridiculously high; nothing could justify a cost of, say, $80 per horse-power for a power plant of the type described, *excluding* electrical equipment.

Here in England the cost of equipping the most modern power plant of 5000 horse-power, *including* electrical equipment, is only £16, say $80 *per kilowatt;* this plant has steam turbines, generators, high-vacuum surface condensers, water-tube boilers with superheaters, overhead coal pockets, coal and ash conveyers, and grateless underfeed stokers; burning a very small coal on these grateless stokers, this station has the record coal cost per kilowatt generated, in regular working at varying loads. CHARLES ERITH.

London, Eng.

I have noticed several comments upon your description of our plant at the Wood Worsted Mill. That signed R. C. W. loses its pertinency when it is explained that the statement that the plates were 7/16 inch thick was wrong and that they are really 9/16 inch. Mr. Lane's assumptions, on page 241, in regard to the number of men employed are entirely incorrect. He speaks of four water tenders; we have none. Each fireman regulates the water supply to the boilers which he fires. He speaks of 32 firemen; we have only eight. Of eight boiler washers; we have one. And eight ashmen; we have two. Of ten coal men; we have five. This is for twenty boilers. I have no apology to offer for my plant as it is and will be glad to discuss its efficiency and its capacity to produce the intended amount of power at the lowest possible cost with any bona fide correspondent, but let me tell how many men I am using and what results I am getting, and don't let them guess at it. Let us discuss my plant as it is, not a straw plant which your correspondents create out of their own minds in order to successfully kick it over. GEORGE H. DIMAN.

Lawrence, Mass.

A Case of Demagnetized Field Magnet

Some months ago our power plant became too small for our business and as we anticipated moving into a larger building in the near future, we decided to tide over by putting in a motor, running on outside current, which in the new plant could be used for a generator.

We bought a 50-kilowatt Fort Wayne 225-volt generator and used it for a number of months as a motor, giving very satisfactory results. When we moved, this machine had to be changed over to generate at 115 volts so as to operate the motors that were already in use. This was done and while the loads were light everything worked beautifully; but one noon, our old engineer, who had just come from the old plant, shut down by throwing off the load by the switches and throwing in rheostat resistance before shutting down the engine.

Everything shut down nicely but when the engine was started at 12:30 o'clock the dynamo would not generate and no amount of persuasion on our part brought any encouragement from the machine, until I got in an expert who threw a light fuse wire across the terminals and started to coax a current through the armature with a piece of copper wire on the commutator. After about ten minutes of this work the machine picked up and worked very satisfactorily for several weeks by letting the generator die down with the engine without throwing off any of the circuits. One noon the engineer forgot himself and threw off the load before shutting down, and on starting again we were up against the same old trouble but overcame it this time by connecting eight battery cells to the field-coils, which made the machine pick up again after about five minutes of running.

Can any of your readers, explain this trouble? I have asked a number of men who know considerable about electrical matters but their explanations are very vague. WM. S. DOUGHTEN.

Philadelphia, Penn.

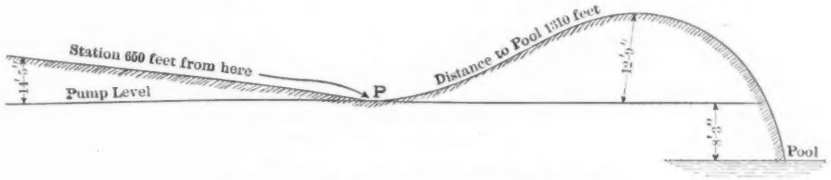
[The failure of Mr. Doughten's generator to "pick up" was of course due to the

loss of residual magnetism in the field-magnet cores and yoke. The cause of this complete demagnetization is well known to those versed in the higher branches of electrical science, and we should like our readers to attempt its analysis. It is not a matter of common knowledge, so no one need feel humiliated if his opinion should be incorrect.—EDITORS.]

"Advice on Pump Location"

In reply to John B. Ash's query in February POWER, page 117, Mr. Ash may rest assured that a pump piped as shown in his sketch will not work, even were it at sea level, and owing to the altitude of Lawrenceburg the conditions are still more adverse. The distance from the surface of the water to the top of the hill is 21 feet and at least 3 inches more should be allowed, as the water will have to be raised to the top of the suction pipe to insure working conditions.

At sea level under the most favorable conditions there is pressure only of 14.7 pounds per square inch. In raising the water 21 feet 3 inches, 9.2 pounds of the 14.7 pounds is used, so there is only about 5.5 pounds pressure left to overcome the friction of the water passing through 1960 feet of pipe to the pump. Under normal conditions a pump to handle about 60 gallons of water per minute will require a suction pipe 3 inches in diameter. The loss in pressure per square inch due to friction in 3-inch pipe discharging 60 gallons per minute is about half a pound per square inch of area for each 100 feet of length, so 1960 feet of pipe would require about 10 pounds pressure to overcome the friction, and there is only about 5½ pounds available for the purpose.



PROFILE OF SUCTION AND DELIVERY LINE

The average pump working under normal conditions at sea level will produce about 75 per cent. of an absolute vacuum; this would be equivalent to about 25 feet of water, and a pump must be in perfect condition to do this. The friction loss in the pipe could be overcome to a great extent by using a pipe several sizes larger than would be required under normal conditions. In this instance 4½-inch pipe would be required.

A pump set and piped as described below will give perfect satisfaction, one having no outside moving parts being best adapted to the purpose. It can be oiled and started and stopped from the main engine-room, and, if pumping clean water, it will run for months without attention.

Set the pump as close to the water as possible and have a foot-valve on the suction pipe. Use wrought-iron bends instead of the ordinary short-turn elbows. In the steam line there should be eight expansion joints packed with some sort of graphite packing; then they will never stick and one packing will last for years. See that the end of the pipe does not rest or bear on the bottom of the stuffing-box on the expansion joint, as otherwise provision will be made for contraction instead of expansion. Place an expansion joint about 10 feet from the pump and another about 10 feet from the boiler. Put a first-class sight-feed lubricator on the pump steam line and keep the pump well and regularly oiled (about one drop a minute will be sufficient). Before starting the first time, pour a pint of cylinder oil into the pump steam line at the engine-room. Keeping the pump well oiled will save no end of trouble. A pump to handle 60 gallons per minute under the conditions mentioned will require a ¾- or 1-inch steam line. Anyway, the steam line should be one size larger than what the pump fitting calls for, as this will make a fair allowance for friction due to the unusual length of the line. There will be nothing gained by making the line larger than I have suggested, as what is saved in friction will be lost by condensation. Two thousand feet is not a great distance to carry steam; the loss by condensation will not amount to much.

In this case I would not advise a separator or steam trap at the pump; it would be better to sacrifice economy to simplicity.

At the lowest point in the water line, marked *P* on the sketch, put a drain-cock. If there is sand in the water, put a sand trap at *P*, made of a short piece of pipe with a valve in it, and the sand will settle in it.

A pump to supply 60 gallons per minute would have a steam cylinder 6x7, water cylinder 5x7, steam line ¾ inch, suction 3-inch, discharge 2½-inch. Owing to the length of the discharge pipe, 3-inch pipe would be best. The friction loss is much less in the latter size, as compared with the 2½-inch. A. H. HALE.

Denver, Colo.

As a member of the POWER family I take pleasure in trying to help Brother Ash. As soon as I read the words "light plant" and looked over his problem, I asked myself, "Why not install right at the pool an electric pump which could be started and stopped from the power house?" If the light plant is only operated six or seven hours per day, I would install a couple of reserve tanks and draw water from them when the dynamos are shut down.

Every engineer knows that the duplex pump is not very economical and this, coupled with the line condensation, oiling, freezing, becoming air-bound, etc., makes Mr. Ash's problem impractical, in my estimation. It would require a duplex pump with 4x6-inch water cylinders to furnish water for the plant.

I was engineer in an ice plant of several hundred tons capacity, which secured water for condensers from driven wells about one thousand feet from the plant. A large compound duplex pump located at the wells drew the water and forced it over the condensers. This pump vibrated considerably and one day it sprung a-leak on the suction side. Of course it ran away, breaking a piston head and knocking out a cylinder head, besides doing other minor damage. When the pump stopped, the water on the condenser stopped also. This let the discharge pressure from the compressors go "out of sight," blow out several gaskets and start as many more leaks before the compressors were stopped and the valves closed. After installing an electric pump our trouble practically ceased. H. G. S.

Kansas City, Mo.

I would advise Mr. Ash that with the pump placed at *P* the service would be unsatisfactory because of the air pocket that would form at the crest of the hill. If he could carry the suction pipe on a gradual ascent from the pool to the 14 feet 5 inches level, with the pump placed at the highest point it would probably work satisfactorily, provided a foot-valve were used at the pool, as the lift would only be 22 feet 8 inches in addition to the friction, which would be very slight if a pipe one size larger than the suction on the pump were used.

In case the pump were placed at the pool and steam carried to it, then a good-sized receiver could be used to advantage to prevent undue drop in pressure. The plan with the pump located at the boiler house is the best. W. H. ODELL.

Yonkers, N. Y.

In locating the pump at *P*, there should not be any difficulty in operating the pump, provided it is in good condition. According to the figures given the lift would be 21 feet in round numbers. I should not consider this excessive, and pumps have been known to operate with lifts of 20 feet in regular work.

The objection to this method of pumping is the trouble which the condensation of the steam pipe will cause. It will not only be a great source of loss, unless the pipes are well protected, but there would

be danger from freezing in cold weather. There would also be a bother in taking proper care of the pump.

If the pump were placed at the pool, the lift could be regulated to suit the convenience of the operator, but the trouble from condensation would be increased, and the operator would have just so much farther to go when oiling and cleaning the pump. The trouble from freezing would remain the same, regardless of where the pump is placed, unless it is properly protected.

As Mr. Ash has electric current available, a motor-driven pump located at the pool would be an excellent solution of the problem.

If the lighting plant does not operate during the day, it would not be amiss to install a tank of sufficient capacity to store the water necessary to operate the ice plant while the lighting plant is shut down. If the amount of water required is 60 gallons per minute that would mean 3600 gallons per hour. Supposing that the generators were shut down about eight hours during the day, it would require a tank to hold about 29,000 gallons. A tank 20 feet in diameter and 15 feet high would hold this amount of water. During the time the electric-light plant is in operation the pump can be supplying the necessary amount of water for the ice plant and by passing an equal amount sufficient to fill the tank.

A pump having a capacity of 30,000 gallons per eight hours would only have to supply about 62 gallons per minute. An ordinary single-piston power pump would do so if the cylinders were 4½x10-inch and the ordinary speed were obtained. If a pump of sufficient size were to be installed so as to take care of the day demand, as well as to pump the water necessary for the plant when the ice machines are in operation, one having a 7x12-inch cylinder would be of sufficient capacity. Under the circumstances cited, however, the ideal pump would be of the centrifugal type, direct connected to a small motor. Figuring on 65 gallons of water per minute, a centrifugal pump having one suction pipe 1½ inches in diameter, and a 1½-inch discharge pipe, operating at a speed of about 1600 revolutions per minute, would do the work nicely. The power per each 10 feet of lift would be about 1¾ horse-power. Such a pump should be purchased for about $50.

E. PRICE.

Cleveland, Ohio.

Replying to Mr. Ash's inquiry, a pump located as suggested at *P* in his sketch is perfectly practicable with a few modifications, and would get water up to the power-house in spite of the statement to the contrary made by one of the two prominent pump makers to whom the matter was referred.

I have had in some twenty years' experience difficulties to face which were much more serious than that proposed by Mr. Ash, which were surmounted by attention to minute details, such as the class and construction of the pump, its packings, connections, pipe joints, absence of air traps, etc. I have in mind at the moment a duplex pump, successfully working for the past ten years, drawing through a 740-foot length of horizontal pipe with 10 feet vertical suction and discharging 12,000 gallons per hour through a two-mile length of discharge pipe with a 60-foot head.

But to deal with the question before us: The horizontal distance from *P* to the pool is 1310 feet, with a total vertical lift of 21 feet. There would certainly be a considerable amount of friction in this long length of suction, but this can be reduced to a minimum by sinking a pit at *P*, say from 12 to 16 feet below the ground level, and placing the pump at the bottom of the pit, which ought to be bricked all around and covered in, as well as drained by a small hand pump. This would bring the suction inlet to from 4 feet to 7 feet 6 inches below the water level in the pool, so that practically the long length of suction pipe would become a siphon to the pump. The area of the suction pipe should also be larger than is necessary for a short suction, and a good foot-valve should be attached to the pool end of the suction pipe.

The steam pipe from the power house to the pump should also be of large area, to allow for condensation and other losses in a length of 650 feet. If the steam pipe is carefully and thickly covered with asbestos or other suitable non-conducting material, the insulation will prevent the loss of steam being so great as to be prohibitive. This pipe line ought also to be fitted, near the pump, with a steam trap and drain cock. The discharge pipe should have a good-sized air vessel fitted to a tee near the pump to take care of any shock of the pump valves and the joints in the pipe line. As only between 50 and 60 gallons is required to be delivered per minute at the power-house, I would suggest a 6x4x6 vertical or horizontal duplex pump (preferably the former), which should easily deliver this quantity under the circumstances cited.

The steam pipe should be 1¾ inches diameter for one-half the distance to the pump, 1½ inches diameter close to the pump, and then reduced to 1 inch diameter at the pump. The diameter of the suction pipe should be 3½ inches, and it would be of assistance to the pump if the crest of the hill was trenched for the suction pipe, say 6 feet, and the vertical hight reduced by this amount. A tee with a screw-down valve should be provided at the hill's crest for filling the suction pipe in the first instance, or a screwed plug (if perfectly tight) would do instead. The foot-valve ought to retain the water and keep the pipe charged while the pump is at rest. The diameter of the discharge pipe should be 3 inches and fitted with a large air vessel, as suggested. Flanged pipes would be preferable to screwed joints for this work, as they are more easily made tight.

Seattle, Wash. J. CREEN.

In reply to Mr. Ash's query in the February number, I would say that he cannot draw water over that hill by suction, as the high places will fill with air, and it will be impossible to keep it free without a regular air pump with suitable connections at each high point. The distance is too great to run a steam line, I think. As the plant is run in connection with an electrical plant, I would suggest that he operate a motor-driven pump at the pool driven from the light plant; that is, if he needs the water badly enough for that. It would be possible to have a storage tank or pond, so that water can be pumped while the electric plant is running, if they do not have a dry service. A 5x7-inch plunger running 60 feet per minute will supply the necessary amount.

Kansas City, Mo. ED. H. LANE.

The conditions under which John B. Ash wishes to operate his pump are very trying. If the pump be placed at the point *P*, I figure that the absolute suction lift would be 8 feet 3 inches, to which must be added the friction head due to the 1310 feet of suction pipe, which would be very great if a pipe of moderate size were used. The distance over the hill from pump level to pump level need not be considered, except as friction head. If the pump is to operate well, the total suction lift must be kept under 25 feet. A pump will lift higher, but will not do so well or easily.

There is also danger of air filling the highest point of the suction line on the top of the hill, and perhaps blocking the pump; it would at least result in more or less trouble with the pump. The steam loss due to the 650 feet of steam line would be large, and it would not be best to consider putting the pump at the pool for the same reason. Either plan could possibly be made to work, but would be expensive and troublesome after being put in operation.

I understand that they are also operating a lighting plant. In that case the best and cheapest way out of the difficulty would be to place an electrically driven centrifugal pump at the pool.

If the lighting plant is now loaded to such an extent that no extra power is available, it would, in my opinion, be cheaper in the long run to install a small generating unit at the plant especially to drive the pump at the pool. Alternating current would be better adapted to this purpose, as the pump motor would then be of the induction type and require very little care.

SAM S. MURDOCK.

Washington, D. C.

On page 117 of the February issue, Mr. Ash has a pump to locate under circumstances very similar to my own experience a few years ago. After tearing nearly all my hair out and quite exhausting the patience of the "old man," I hit upon the following plan: The suction line was 1680 feet long with a lift of 11 feet 6 inches; the diameter of the pipe was one inch. I placed one tee, with the open leg upward, 500 feet from the pump in the suction line, and another one 500 feet from it in the same position. Then I screwed in a piece of pipe of a length just half the distance the pump had to lift the water and placed a fine screen over the open end to keep out all foreign matter. (The upright pipes must be perpendicular to get the best results.) I used steam to see that all the joints were tight and then filled the line with water.

This line was used 30 days without trouble and the pump was stopped from twelve to fourteen hours in each twenty-four. By lowering the suction line eight feet over the crest of the hill the lift will not be much and the upright pipes act on the same principle in the suction line as they would in a gravity line.

By placing the pump at the pond the steam would drop, in piping it that distance, from 20 to 30 pounds.

Muncie, Ind. G. W. FELVER.

In the February number, on page 117, John B. Ash asks about a pump location, and gives a sketch. I should suggest the use of an electric motor as the best solution of the problem, with the pump and motor at the pool. He can use the generator also for the lights. The loss in the steam pipe would go a long way toward paying for the motor and line in a year.

If, however, he does not care to install a motor, I should suggest the following: Locate the pump at *P*, run the steam main underground inclosed in a tile duct, with asbestos sponge as an insulation. The pipe from *P* to the pool should be absolutely air tight and a perfect foot-valve placed at the suction end. Some method of priming will be necessary.

The piping required will be 4-inch for the suction, and anything from 2-inch to 3-inch for the delivery. The size of pump required will be approximately 4½x4x8, at 80 pounds pressure, with approximately 60 strokes a minute. The loss over the piping should not be extreme if the pipe is well covered, but the cost of the installation would probably be heavy.

By the use of a motor and an automatic starter, either alternating or direct, the pipe line could be reduced to 2-inch. A motor of 6 horse-power connected to a duplex centrifugal pump would do the work at one-half the cost, as there would only be the pipe line from the station to the pump and the pole line, against a 2½-inch line per 650 feet; a 4-inch line per 1310 feet, and the steam piping with insulation per 650 feet.

If Mr. Ash will send me a few more details regarding his lighting plant, I will give him further information.

Boston, Mass. HENRY D. JACKSON.

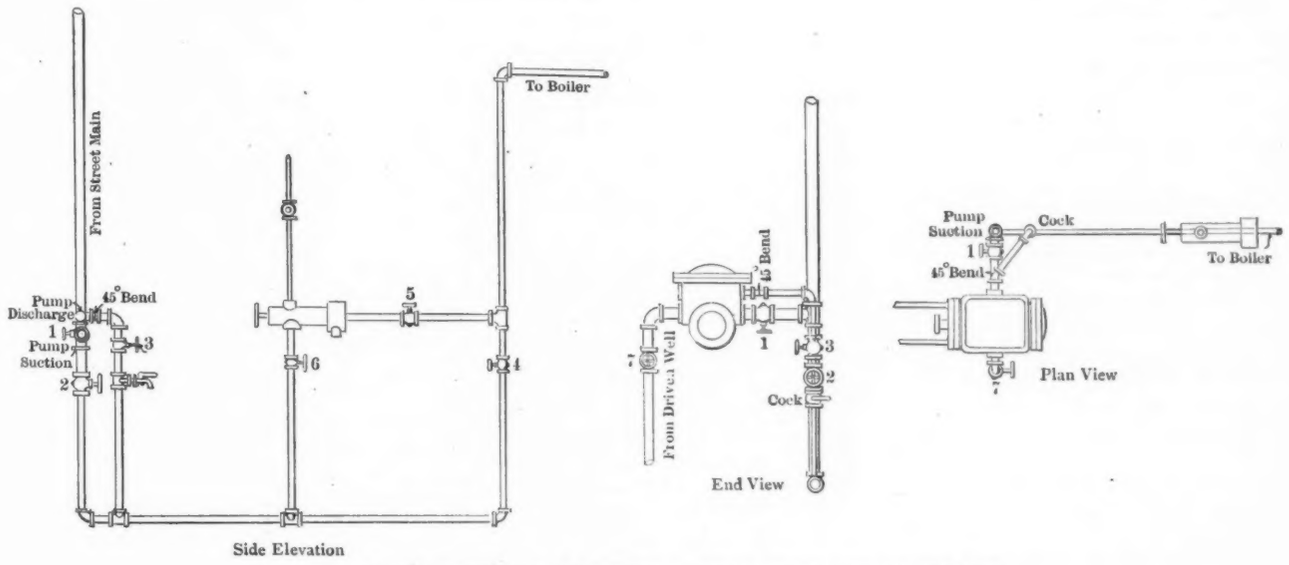
"Piping a Steam Pump and an Injector"

I believe the arrangement of piping submitted herewith to be much better than that planned by W. H. Wakeman, as published in POWER for January and March. In his system no provision is made for drawing water from angle-valve 5 when the pump is put out of service by a breakdown. With my arrangement there is always water on tap.

In this instance I advise the using of straightway valves, with the possible exception of the one in the injector feed-pipe. All valves between the street main and the boiler should be straightway, so as to cause as little delay as possible in filling the boiler after the Sunday wash-out.

I will have to take issue with Mr. Wakeman, too, when he says an "angle-valve will not cause any more friction than an elbow" in the suction pipe of the pump leading to the driven wells. It is admitted by all authorities on the subject that the ordinary short-bend elbow causes a serious amount of back pressure. If this is true of the elbow with its short but comparatively gradual turn when compared to the angle-valve, what must happen when a solid body of water strikes squarely against the bottom of the bend in an angle-valve?

As an illustration of how little inclined a body of water is to suddenly change its course when meeting an obstacle, I will call his attention to the breaking of heavy iron fittings on pipe lines when a



A. H. HALE'S SUGGESTION FOR PIPING A STEAM PUMP AND AN INJECTOR

valve is opened too quickly. In the suction pipes of pumps using valves the water has an intermittent flow caused by the opening and closing of the suction valves. Now, anything in the suction pipe that interferes with the free flow of the water to the pump will increase this pulsation of the water, and in some instances I have known the pumps to fail to fill on account of it. Of course the pumps pounded badly. In the suction pipe of this pump a wrought-iron bend instead of the short-bend elbow would be best, and a straightway valve in place of the angle-valve. In fact, bends would be much better than elbows whenever the latter are called for.

To operate the system of piping as shown by me, starting with all valves closed in each instance: To pump from the street main into the boiler, open valves 1, 3 and 4. To pump from driven wells into the boiler, open valves 7, 3 and 4.

To fill the boiler from the street main, open valves 2 and 4. A. H. HALE.

Denver, Colo.

As my article under the above title in the January number received some attention in March POWER, I wish to offer the following facts and ideas for further consideration along this line: It is claimed by "Scotty" that it takes only from one to five seconds more to open a gate than a globe or an angle-valve. This is one of those cases where the extra time actually required seems much more than it really is, but it proves annoying and troublesome in direct proportion to its seeming. Furthermore the difference will be more than above-mentioned with some kinds of gate valves. In a plant that I formerly had charge of, there was a 5-inch gate valve that required thirty-one revolutions of the wheel to open, and such useless trouble ought to be avoided as far as possible.

Every morning that my present plant is started, in winter, it is necessary to open or close more than one hundred valves of different kinds, and if they were all gate valves it would cause waste of time, when there is none to spare, and prove very exasperating.

I am told that if cross-valves become unfit for use it costs more to replace them than it would if other kinds were used, but I have not found it so in practice. At present a cross-valve costs but five cents more than a gate valve of equal quality, therefore the difference is not worthy of consideration.

Very few valves are so ruptured in service as to make it necessary to replace them—but they do get leaky more or less. I have found many gate valves that leak when new, or after they have been used but a few months, and as a rule it is not only expensive, but impractical to repair them; hence it is cheaper to throw them out and put in others that can be repaired.

If a globe, angle- or cross-valve leaks, a new disk will frequently prove a remedy. If not, it is only necessary to put a valve file in place, give it a few turns, and the seat is perfect, making it a practically new valve.

The following comparison will prove interesting: An 8-inch gate valve leaked badly, and was repaired by an expert mechanic, but it took twelve hours of honest labor to make it tight. A 6-inch angle-valve in the same plant leaked excessively. The bonnet was taken off, the old disk removed and a new one inserted, making the valve perfectly tight. The time required for the latter job was one hour, to which must be added the cost of a disk, which is about equal to another hour, making quite a difference in the cost of repairs.

Concerning the comparative cost of globe and cross-valves, I would say that the former is about ten per cent. cheaper than the latter, but when a globe valve is used, a tee must be purchased, and that makes the total equal the cost of a cross-valve, as a tee is not required in the latter case.

"Scotty" has something to say about the comparative friction of globe and angle-valves, but there are so many things that can be slightly different when tests are made, that unless they are all carefully taken into consideration the tests are of little value. Is he sure that the water pressure in the discharge pipe was the same for all of his tests? He says that the friction of an angle-valve equals that of a globe valve, but the question to be considered is, does the friction of an angle-valve equal the total friction caused by a globe valve and an ell?

My plan calls for cutting forty-two pipe threads, while it is necessary to cut fifty-one for the plan that "Scotty" favors. If a steam fitter should take a contract for a job of piping, having made his price according to certain conditions, and when the time arrived for the work to be done he should find that somebody had substituted another plan calling for twenty-two per cent. more pipe threads, he would probably object strongly to the unnecessary expense, and his objection would be sustained.

Special attention is called to the fact that I am not writing in favor of gate valves, as I never use them if it is practicable to use globe, angle-, or cross-valves instead. W. H. WAKEMAN.

New Haven, Conn.

Referring to Mr. Wakeman's article in January POWER on this subject, I would like to ask if a pump or an injector delivering water to a boiler would work correctly if a street-main supply was feeding into the same boiler, and at the same time as the said pump or injector, as shown in Mr. Wakeman's sketch on page 31. W. H. TYLER.

Lowell, Mass.

The above letter was submitted to Mr. Wakeman, who replied to it as follows:

Replying to Mr. Tyler's inquiry, I would say that with the arrangement shown on page 31 of the January number, valves 7 and 8 may be opened, and the injector started in the usual way. Water will then be fed from the street main direct, and also from the injector at the same time. I tried this experiment less than a week ago.

It is necessary to observe three points in this connection: First, the street-main pressure must exceed the boiler pressure; second, the piping and valves must be large enough to easily convey the two streams of water, and third, the injector must be in perfect order.

I have tried this with an injector that was not in perfect order, and it would not work, although the injector would feed the boiler alone. After putting the injector in perfect order, it would feed in connection with the street-main service, or in company with other boiler feeders.

The experiment that I tried less than a week ago was made with a Hancock inspirator 13 years old.

W. H. WAKEMAN.

New Haven, Conn.

Why Not Do Away with Screw Propellers?

The determination of screw-propeller proportions suitable for the high speed of the steam turbine is being subjected to extensive investigation, both in this country and abroad. The difficulty lies in the fact that the propeller has its highest efficiency at a relatively low speed of rotation, while the steam turbine has its best efficiency at high rotative speed; and it has been proposed to make a compromise by raising the speed of the propeller as much as possible, at the same time decreasing the speed of the turbine. A propeller, however, cannot come up to a speed that suits the turbine, and the turbine cannot be cut down in speed to suit a propeller. It would seem obvious, therefore, that no compromise is possible. But why not entirely dispense with the screw propeller in connection with steam turbines and find some other means for marine propulsion?

By increasing the revolutions per minute of a propeller, during recent experiments, it was soon found necessary to provide means to prevent the water from leaving the blades sidewise, so the propeller was put inside of a short cylindrical housing, where it was run with very small clearance. Further, it was deemed advisable to lead the water to and from the blades in the proper way and for this reason the housing was provided with guide-blades, both in front and back of the propeller. Next it has been proposed to put these parts inside of the hull in a space provided therefore (see *Die Turbine* for October, 1906), and in such case, for the sake of doing away with a long shaft, it would be just as well to place the propeller amidships near the turbine, and provide proper channels or tunnels for the inlet and outlet of the water.

We are now at the point I am driving at, for the next thing would be to substitute an *impeller* for the propeller, which would have the advantage of permitting the turbine to run at the most efficient speed. In other words, why not jet propulsion instead of screw propulsion?

There might be another way to achieve the same end without sacrificing the screw propeller, viz., to design the ship's machinery as a regular power plant, with high-speed turbines direct-connected to high-tension alternating generators, whose current, properly transformed, could be transmitted to the slow-running

propeller shafts. Both turbines and propellers would in this manner work to the best advantage and the problem of reversing would be very easily solved.

Boston, Mass. MICHAEL NYROP.

"A Question in Belt Operation"

Becoming interested in "A Question in Belt Operation," raised by R. T. Strohm in the November issue, and having read with increasing interest the answers in the December and January issues, I would like to add one more thought to the discussion.

A belt will always endeavor to run in a direction at right angles to its line of first contact with the face of the pulley, as in Fig. 1. The same explanation applies to the crowned pulley, as shown by Fig. 2, but with a different result. In the first case there is but one angle for the full width of the belt, and the belt runs to one, or the so-called "low," side. But in the second case there are two angles, counterbalancing each other; consequently, the belt runs centrally. If the crowned pulley is out of line, however, the result will be the same as in the first case, as the two angles will be out of balance.

North Adams, Mass. W. J. P.

In recent issues of POWER I have noticed a discussion as to which side of a pulley the belt will run off when the shafts are not in line. Judging from the correspondence so far published, I think there exists a slight misunderstanding of the subject, which would tend to leave a casual reader in a state of doubt. Those writers who take the "high-side" stand cite the case of a "crowned" pulley, which will cause the belt to climb to the crown, provided the shafts are parallel and level. A belt will not remain on a crowned pulley any more than on a pulley not crowned, if the pulleys are revolving in different planes, without the redeeming feature of their being (in relation to the lead of the belt on the pulleys) in line. "That's where the rub comes in," I think.

When the shafts on which the pulleys are mounted are parallel, level and, in short, in line and "square," as we say in shop talk, the belt will remain on no matter which is the driving pulley, or in which direction the pulleys revolve. Nor does it matter (at least, not very much) whether the pulleys are crowned or otherwise. But if the shafts are not parallel, even though both be exactly level, the belt will run off the so-called "low" side of the pulley (Fig. 1), whether crowned or otherwise. The reason is, the pulleys are revolving in different planes; each plane tending to run the belt in its own direction, toward that end where the shafts are closer together. In the case illustrated it makes no difference which is the driven, or in which direction the belt runs.

Of course, if the shafts are not out very much, the belt will probably remain on; but there is a limit for each and every case, which cannot be exceeded, else the belt will surely run off. That the belt does not run off in any given case shows that the shafts are not out enough, instead of proving that the belt will not run off at all in some other case.

If a pair of shafts with a pulley on each are parallel and level, and one of the pulleys is cone-shaped, the belt will not run off, as some seem to think, simply because the pulleys are running in the same plane. In Fig. 2 the shafts are parallel and level. The belt will remain on, but will probably run to the "high" side of the conical pulley, and overlap or overhang the edge by a certain amount, which can only be determined by actual experiment. The conical-pulley case referred to is not at all like that where the shafts are "out of parallel," although some persons have an idea that it is.

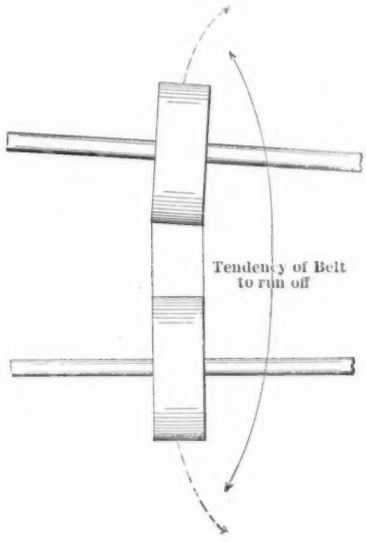


FIG. 1 AS MR. MASON VIEWS IT

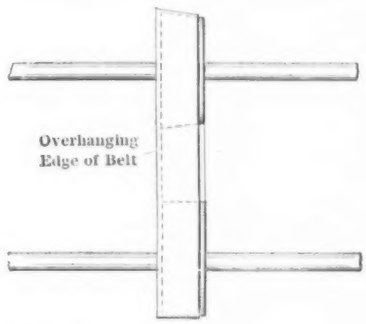


FIG. 2

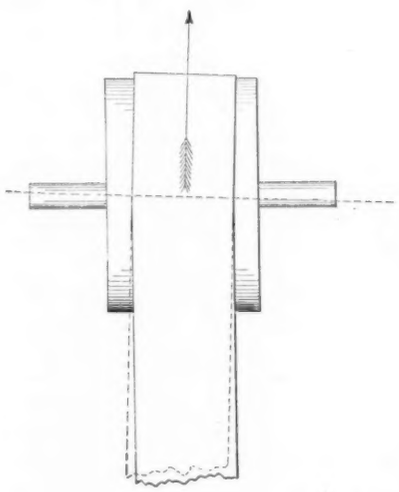


FIG. 1 W. J. P.'S EXPLANATION

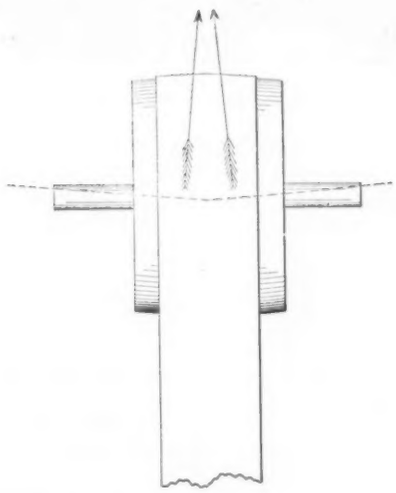


FIG. 2

In a case where the shafts are in parallel planes, but are not level, the belt will remain on the pulleys; for in this case it approaches the conditions obtaining in quarter-turn or twist-belt drives. Here, also, there are certain limitations, unless *all* the conditions upon which quarter-belt drives depend are fulfilled.

The things to be considered in such questions are the planes in which the pulleys revolve, and how the belt leads to the face of each pulley. If the leads are direct, fair or straight (or any other way you like to state it), the belt will stay upon the pulleys, even though they do not run in the same plane, as in the case of a quarter-turn belt. But if the belt does not lead on the pulley correctly, then with pulleys running in different planes, it—the belt—will not remain on.

I think it would be instructive to have a model made that would illustrate approximately all the various cases that can happen in practice in relation to belts and pulleys, with shafts in and out of line. From such a model could be seen just what will occur in a given case.

Scranton, Penn. CHAS. J. MASON.

Lap-seam Boilers—Test Pieces from the Lynn Explosion

In my report of the boiler explosion at Lynn, Mass., published in February

POWER, I mentioned that tests were to be made of the plates as near the original fracture as possible, to test their ductility and general fitness for use in a boiler. In tests that I had previously caused to be made, of other exploded boilers, the results showed as far as these tests could that the steel was of good material, and yet failed, with all the appearance of brittle structure. The tests of the Lynn boiler, which were made under the direction of Chief Inspector McNeill, some being made at the Watertown arsenal and a number at the Institute of Technology, agreed substantially.

Following is a report of two of the tests:

	Lengthwise of Grain.	Crosswise of Grain.
Area of original section, sq.in.	0.6435	0.5267
Elastic limit, lb. per sq.in.	31,100	28,500
Maximum load, lb. per sq.in., original section	61,200	55,100
Area of fractured section, sq.in.	0.345	0.271
Reduction of area of cross section, per cent.	46.4	48.6
Ultimate extension, per cent. of 8 in.	19.2	21.5

Chemical test, Phosphorus, 0.055; sulphur, 0.040.

From this it will be seen that the results indicated in both tests that the steel would ordinarily be passed as a safe steel, even if in some instances it might be slightly improved.

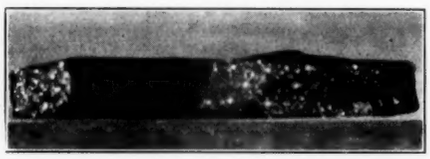
Physical tests were also made. Test pieces were made and bent double, cold. Some of these had planed edges, while the others were identical, except that the edges were unplaned and left as they came from the shears. In each case the planed specimens bent double without showing any fracture at the bend, while the unplaned pieces showed fractures at the bend immediately after the bending commenced. The latter indicated all the characteristics of a brittle steel, while the planed pieces from the same sheet showed every indication of a most ductile steel. This would seem to emphasize the important influence, on the behavior of the steel in the boiler, of shearing, punching holes, etc., the starting of any minute cracks by these operations, and the importance of eliminating these cracks by afterward planing edges or drilling out the punched holes.

These things doubtless have been observed before by boiler designers, and may possibly have little influence on the lap-seam failures, except in so far as the shocks delivered to the plate by shearing and punching and manufacture may affect the steel adversely for a short distance from the place where the blows were delivered. It is a well-known fact that shocks to steel, many times repeated, have had the effect to crystallize the steel and give it a brittle character. A piece of steel from this exploded boiler, which I broke off from the fractured plate with a single blow of a hammer, has nothing about it that would lead one to expect so good a test as above reported, and it merely indicates how complex the problem is, as to just what changes do take place in the steel, where these changes are located, and how to best deal with them.

As an outcome of the disaster, it seems quite certain that the future installation of lap-seam boilers over 40 inches in diameter will be prohibited in Massachusetts, and factors of safety adopted for the remaining boilers that will quickly retire them from use.

I recently noticed an opinion that the expansion and contraction of boilers set in brick-work with careless handling is the cause of these cracks appearing, but I have seen many of them in large upright boilers, which were not subject to the severe strains the improperly cared for return-tubular boiler may receive.

I have observed, also, an opinion that there are not many boilers that have these cracks, anyway, that they are the exception and really quite rare; yet I took the trouble at one time to look this matter up by having parties who are engaged in the business of cutting up old boilers take note of this matter, and they found that over one-half of the seams they cut open had lap cracks pronounced enough to be very noticeable, while there was no doubt



FRAGMENT FROM LYNN BOILER

that many had small lap cracks that did not meet their attention. The lap-seam danger is a real one, but modern high pressures and the forcing of boilers is likely to bring it to light more sharply than was the case years ago.

Boston, Mass. THOS. HAWLEY.

Large Boilers—Compounding an Engine—Power Required to Drive Cotton-mill Machinery

On page 784, December POWER, a correspondent who signs himself "Lancashire" wants to know a few things.

I do not find anything unusual in the boiler being 32 feet long and 9 feet in diameter, although it is admittedly a little above standard size for the modern high pressures. Lancashire or double-flued boilers are constructed for standard sizes from 9 feet to 30 feet long, and from 6 feet 6 inches to 8 feet 6 inches in diameter, for a working pressure of 110 pounds per square inch, and with flues 39 inches in diameter in the larger boiler, with 6 feet length of grate, 39 square feet of grate surface and an effective heating surface of 900 square feet.

The writer has inspected the boilers and machinery of many mills in both Lancashire and Yorkshire, as well as in the Midlands, and has come across Lancashire boilers 37½ feet, and up to 42 feet long, although I believe these lengths are obsolete now, so far as new construction is concerned.

It is not unusual to compound an engine, especially when installing a new boiler of a higher working pressure. There are two elements that compose the absorption of power by the engine and gearing: friction due to the dead weight of the working parts, piston and rod packings, etc., which constitutes the initial or slow-speed friction of the engines; and the friction of the engine due to the working load.

It has been found that taking the average of a large number of cases, 0.137 of the indicated horse-power was required to run the engine. Therefore, as the indicated horse-power is given at 250, the power required to run the engine at 50 revolutions per minute, with all the machines thrown off, would be 34¼ horse power.

For "Lancashire's" further information I append the power required to drive the various machines in a cotton mill, although this may vary somewhat with the counts of yarn to be spun and the lengths of the various machines used in the card and spinning rooms. The horse-power, however, may be taken as about the average for each machine:

	Horse-power
Double scutcher	3
Single scutcher	2
Single vertical opener	2
Double carding engine	1½
Single carding engine	⅞
Drawing frame	½
Slubbing frame (60 spindles)	1
Intermediate frame (80 spindles)	1
Proving frame (100 spindles)	1
120 ring spindles	1
60 ring spindles (including preparation machinery)	1
75 mule spindles (including preparation machinery)	1
250 mule spindles	1
Pirn winder (160 spindles)	1
Drum winder (200 spindles)	1
Sectional beamer	1
Bobbin (cop) winder (300 spindles)	1½
Self-stopping beamer (3 machines)	1
Size mixer (single) with pump	1½
Size mixer (double) with pump	2¼
Sizing machine (slasher)	4½
Plain looms, 40-inch reed space, 180 picks, 6 per	1
Box drop looms, 40-inch reed space, 150 picks, 4 per	1
Fustian looms, 40-inch reed space, 145 picks, 2½ per	1
Double left dobbies, 20 shafts, 9 machines, per measuring and folding machines	½
Hydraulic cloth press (2) 1¼-inch rams	3½

This also includes 10 per cent. allowance for frictional resistance of shafting.

I was not a little surprised to learn that the man who called himself "the engineer" was totally uneducated; such a man in responsible charge of engines and boilers in England is assuredly a *rara avis;* and if my memory serves me I think the law

has made it compulsory for some years past for every man in charge of steam boilers to have a certificate of competency.

Seattle, Wash. J. CREEN.

Mr. Lincoln's Problem in Transformer Connections

I submit herewith a diagram in solution of E. S. Lincoln's problem in February POWER. In Fig. 1, *A* and *B* are three-pole double-throw switches used to connect the transformer in star or delta. When the switches are up the transformers are connected in star; and when down, in delta. Two three-pole single-throw switches are mounted at *C* so that the lower switch blades may enter the clips of the upper switch, row *1* being the blades of the upper switch and row *2* the corresponding clips, while *3* and *4* represent the blades and clips, respectively, of the lower switch.

To connect the transformer windings in parallel both switches are turned down, connecting *1—2* and *3—4*. To connect in series, the lower switch blades are turned up into the clips of the upper switches, thereby connecting *2* and *3*. While in this position it is impossible to make a short-circuit by closing the other switch.

In Fig. 2, *D* is a three-pole double-throw switch and *E* is a three-pole single-throw switch. To connect the windings in parallel both switches are turned up. To connect them in series, switch *E* is opened and switch *D* is turned down.

In Fig. 2, it is possible to make a short-circuit by leaving switch *E* in when switch *D* is turned down; therefore this connection is not quite up to requirements.

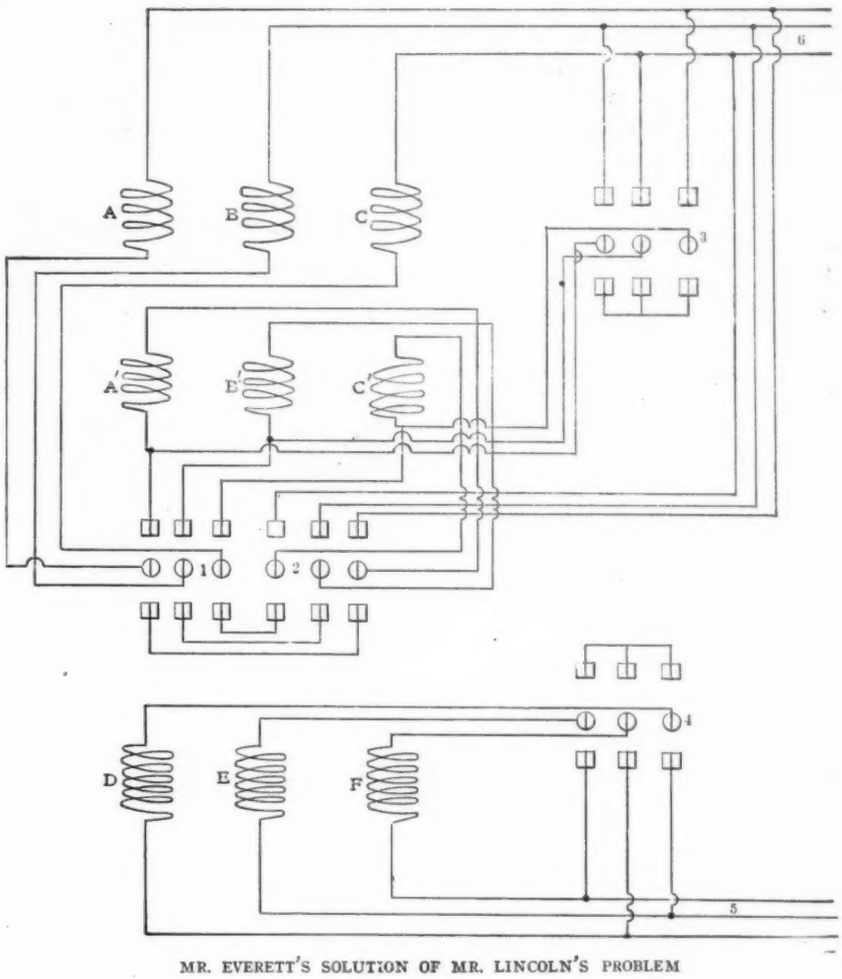
In a transformer to deliver 110 volts, it would require a ratio of 9.454 to 1 with a primary voltage of 2080. If the primary voltage was 2200 the ratio would be 10 to 1.

CHAS. O. RANKIN.

Craftonville, Cal.

The accompanying diagram is my idea of the proper connections for the solution of the transformer problem in the February number.

A and *A'* are the secondary coils of one



MR. EVERETT'S SOLUTION OF MR. LINCOLN'S PROBLEM

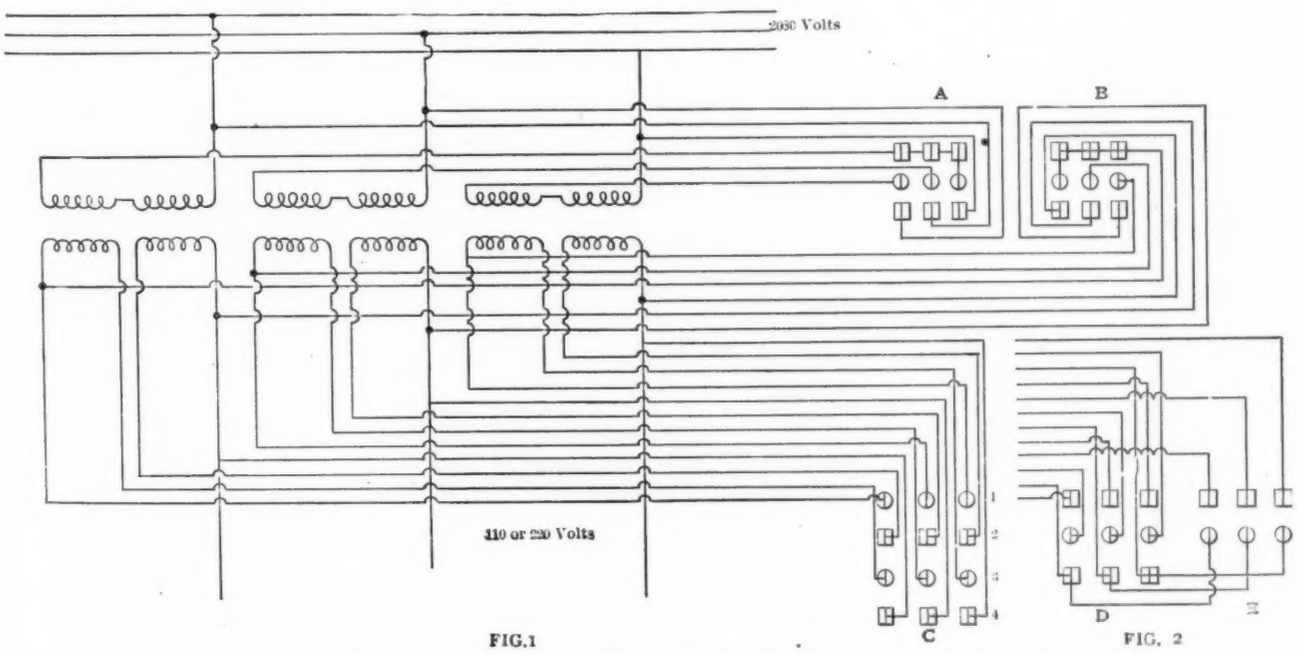


FIG. 1 FIG. 2

MR. RANKIN'S SOLUTION OF MR. LINCOLN'S PROBLEM IN TRANSFORMER CONNECTIONS

transformer, *B B'* and *C C'* of the others; *D, E* and *F* are the primary coils; as the two primary coils of each transformer are always in series they are not shown separately; *1, 2, 3* and *4* indicate the switches, all of the three-pole double-throw type; *5* represents the primary mains and *6* the secondary mains.

When *1* and *2* are up, the secondary coils of each transformer are in parallel; when *1* and *2* are down, the secondary coils of each transformer are in series. When one of these switches is up and the other down, one coil of each transformer will be connected properly and will deliver 110 volts, while the circuit through the other will be open.

When switch *3* is up, the secondary coils are connected in delta and when *3* is down they are connected star fashion. Switch *4* performs the same function for the primaries, but it is thrown down to connect in delta and up to connect in star.

HARRY D. EVERETT.

Jamestown, N. Y.

"The Centrifugal Pump"

The February issue contains, on page 88, an interesting article on "The Centrifugal Pump." This class of pump has its field, but it is restricted in extent. Its limitations are soon reached. For all-round work the common wasteful steam pump still holds its own. With the steam pump

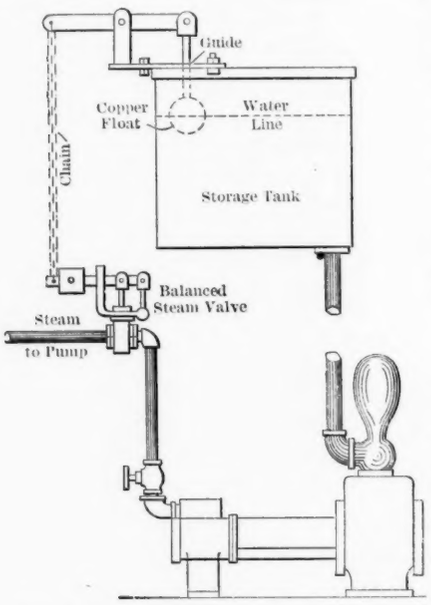


FIG. 1. METHOD OF GOVERNING BY FLOAT AND BALANCED STEAM VALVE

the engineer can control the speed, and consequently the output, to suit the demand. If it is to keep a certain tank full, a float in the tank connecting with a balanced valve in the steam line to the pump offers an ideal regulating device, as shown in Fig. 1. If the function of the pump is to keep a sump or tank from overflowing, the float is set near the bottom of the supply tank and connected to the balanced valve in such a manner that when the tank is nearly empty the pump is shut off automatically and again started up upon water flowing into the tank.

For boiler or similar service, when the pump has to maintain a certain pressure in discharge, the output may be controlled

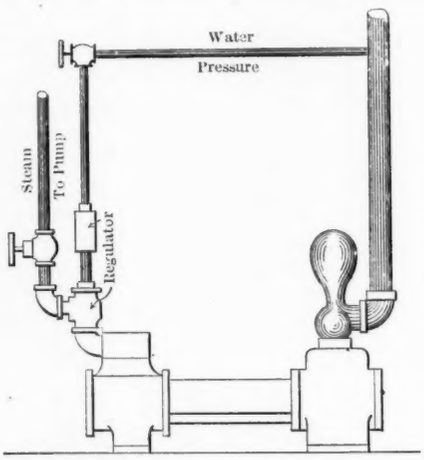


FIG. 2. METHOD OF GOVERNING BY PUMP-PRESSURE REGULATOR

by the well-known method of using a pump governor, which operates a balanced valve in the steam line to the pump by pressure from the pump discharge, as shown in Fig. 2.

With the centrifugal pump I know of no workable controlling device. I have tried, unsuccessfully, to regulate the output of a No. 10 centrifugal pump by throttling the discharge, automatically and otherwise. Even if the pump is direct-connected to an engine or motor, reducing the speed is impracticable, because reduction of speed brings about reduction of head, and this soon stops the flow.

While the first cost of a centrifugal pump is less, much less, than its steam competitor, yet when the cost of countershaft and belting is added, the cost of the steam pump is not greatly in excess of the centrifugal.

Your correspondent mentions cylinder-oil as an item of cost not found in the centrifugal pump. Does not the cost of maintenance of belting for the centrifugal pump more than offset the oil bill for the steam pump?

Further referring to the centrifugal pump, he says, "The only wearing parts are the bearings." He has probably never had a hurry call to a centrifugal pump which was apparently working, yet failed to pump, and, taking the pump apart, found the shaft in the back end of the stuffing-box reduced to about one-quarter its original size and twisted off. Nor can he have had to renew his pump blades every few months, on account of the erosion caused by the blades churning through milk of lime or other gritty matter at the rate of one thousand revolutions per minute. Under these conditions the average life of a shaft for a No. 3 pump is four months under continual service, and about seven months for the blades.

It is presumed that if these pumps were two-stage, instead of single, better results would be obtained. Still, I have more respect for the centrifugal pump than for reciprocating pumps. The rotary motion and symmetrical appearance of the former seem to come nearer the ideal.

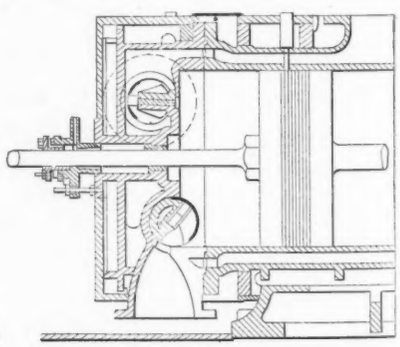
Glendale, Ariz. W. T.

Cylinder and Piston Wear

I was much interested in the article by L. H. Edwards in the December issue on cylinder and piston wear. It is noticeable that such wear occurs especially in the large cylinders of compound engines, on which the condenser vacuum is exerted. Consequently it occurs to me that it may be due in part to the fact that this vacuum removes from the cylinder all trace of moisture, which would help to lubricate it, as well as the oil deposited on the sides; because it will be found that these cylinders wear also at the head end, where there is no ring pressure.

The wear at the head end, however, has been attributed chiefly to the plan of reaming the cylinder with only slight feed of the tool per revolution, instead of making a finishing cut with a feed of from 8 to 10 millimeters per revolution. The last method is preferable for the reason that it leaves but few asperities, so slight as to be readily smoothed by the rings during the first revolutions of the engine. It would be interesting to know if there are engine builders who scrape the cylinders, as for a plane surface.

I have noted that pistons having a rod coming out of the back junk ring seldom wear. This is the design adopted by McIntosh & Seymour. Rice & Sargent babbitt their pistons, but the rod does not



CYLINDER AND PISTON-ROD OF FARCOT ENGINE

traverse the junk ring. They have, however, a speed of 4.80 meters a second. Allan metal has been applied to many pistons with good results.

Farcot, the great French engine builder, built only engines with one cylinder, which did not wear, in spite of the large area beyond the rings. He used a number of

packing rings which retained oil and easily lubricated the parts of the cylinder where the vacuum had removed the moisture. I submit herewith a blueprint showing a Farcot engine cylinder and piston.

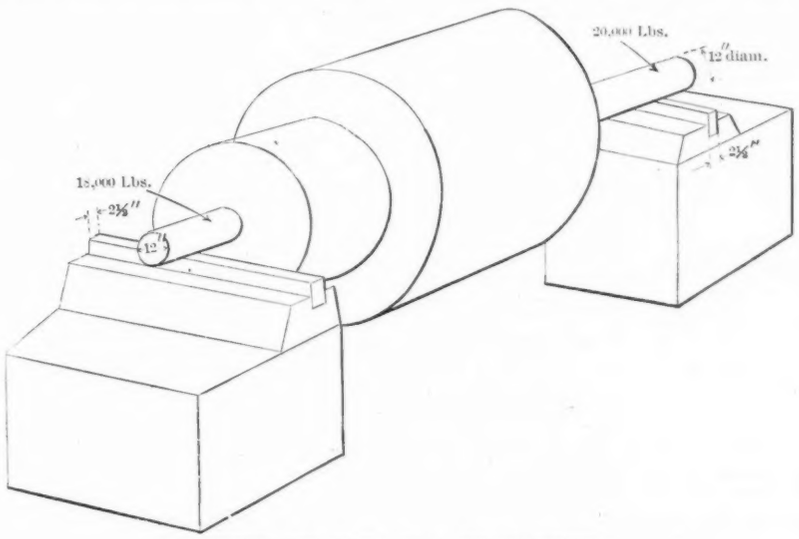
I should like to hear from other engine builders on this subject.

Brussels. H. BOLLINCKX.

Balancing the Rotor of a Steam Turbine

It is desired to balance a rotor for a steam turbine on parallel strips 2½ inches wide, the journals being 12 inches in diameter and the distribution of the weight of the rotor such as to give a maximum load of 20,000 pounds at one end and 18,000 pounds at the other, as indicated in the accompanying sketch.

Is the width of 2½ inches sufficient for this weight to prevent the journals from being bruised, scored or otherwise injured during the balancing?



BALANCING THE ROTOR OF A STEAM TURBINE

Can any POWER reader give me a rule for properly proportioning the tread for balancing fixtures from the information given herewith? G. J. WALTON.

Frankford, Philadelphia, Penn.

"Automatic Engine-stops versus Fly-wheel Insurance"

Referring to a recent controversy, I fail to see where the engine-stop people and the liability companies have anything to argue about. If a man builds a building he insures it at once. If he has a good sprinkler system put in, he gets a lower insurance rate and everybody feels better all around. Of course, if it came to choosing between the two there might be some room for an argument, but it is just like the safety valve and the boiler insurance—we need them both.

Kansas City, Mo. ED. H. LANE.

"Why the Difference in the Steam Line?"

Regarding Mr. Bach's diagrams on page 115 of the February number, and his inquiry, "Why does the steam line fall in No. 1?" I would say that the No. 1 diagram is the natural low-pressure diagram showing the normal pressure of admission, and consequently the normal steam line, which falls naturally as the volume of steam between the low-pressure piston and the intermediate valve becomes larger as the piston advances.

In the No. 2 card the same would have held good, but in assembling the indicator the spring was so adjusted with regard to the piston length that the travel was restricted, and the pencil could not show the true pressure until cut-off was nearly reached, and the pressure had fallen to where the indicator could record it. It will be noticed that the two diagrams are nearly alike after cut-off takes place.

Mansfield, Ohio. BURTON B. PIERCE.

"Boiler Furnaces and Combustion"

I have been much interested in the discussion between Messrs. Holbrook and Van Brock on "Boiler Furnaces and Combustion." A few years ago my company installed automatic stokers for two reasons: first, they hoped to secure the elimination of smoke; second, they anticipated increased economy. The first object we had no trouble in accomplishing, while the second was about on a par with hand-firing. For various reasons, however, we took the stokers out. Then, after several experiments, I decided that with sufficient depth of furnace I could get practically smokeless combustion.

The greatest depth possible without raising the boilers was 36 inches between the boiler and grate, at the front, but pitching the grates 8 inches gave me a depth of 44 inches at the bridgewall and 40 inches at the center of the furnace. I would have built a "dutch oven" if I had had the room, but I am satisfied as it is. We increased the capacity 20 per cent., have no difficulty in burning 25 pounds of coal an hour per square foot of grate surface, and have secured an increased economy of 1.5 per cent. This, I think, is proof enough that to burn bituminous coal properly it must have room.

A setting such as Mr. Van Brock advocates may be suitable for anthracite, but not for bituminous coal, and I believe that it is the recognition of this fact which has enabled the stoker engineers to make such progress in smoke prevention, as compared with conventional hand-fired furnaces in which the grates are only 24 inches from the boilers.

Gloucester, Mass. P. A. LEAVITT.

Wants to Know About Motor Drive

I have a large lathe, a drill press and several other machines to install in our shop, and I don't know whether to have them motor-driven or not. The lathe is a special one for facing pipe flanges, etc., and would require a 3-horse-power motor; the drill press is an ordinary 3½-foot radial, and would require a 2½-horse-power motor. I have also a 20-inch pipe machine which will require one 5-horse-power motor. All the motors would have to be of the variable-speed, reversible type, to operate on a 500-volt direct-current supply circuit.

I wish some POWER reader would help me to decide whether it is practical to install motors on these machines for individual drive.

Kansas City, Mo. H. G. S.

Should Lignite be Wet or Burned Dry?

Lignite is used at our station as fuel, and one of our firemen burns it dry, while the other insists upon wetting it. Which is the better practice, and why?

JOHN M. MACGREGOR.

San Antonio, Tex.

Magnet windings of uninsulated wire are said to have proved feasible by the use of aluminum wire, the natural oxide upon which forms an effective insulation for moderate voltages. For over 200 volts, paper wound wet between the layers is effective; and for higher potentials, extra oxidization has to be secured by dipping in a chemical bath.—*Model Engineer and Electrician.*

POWER

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Horizontal versus Water-tube Boilers and Manual versus Mechanical Coal Handling

The description of the steam plant of the Wood Worsted Mill, which appeared in our February issue, has occasioned, as was anticipated, a lively discussion as to the comparative net or final efficiency of the simpler apparatus and methods employed, and that obtainable with a more elaborate installation. Mr. Diman replies to some of his critics on page 250 of the present issue, and, too late for use in the correspondence department, sends additional information. Mr. Lane on page 241 writes of water-tube boilers which have been in use in Kansas City since 1887 and thinks that Mr. Diman's horizontal tubulars, even if lower in first cost, will cost as much or more per horse-power-year when their shorter life is considered. Mr. Diman does not admit that the horizontal return-tubular boiler is limited to a shorter service than the water-tube type. He cites horizontals under his charge at the Washington Mills, which were put in also in 1887, and which have been habitually run at fully 50 per cent. over their rating. Last year the pressure allowed on these boilers was reduced to 125 pounds on account of age. They have been placed upon other work for the same company at that pressure, and Mr. Diman sees no reason why they are not good for another twenty years' run.

"The boilers at the Wood Mill," says Mr. Diman, "were put in for 200 horse-power each. On March 7 nineteen of these boilers evaporated between 6:30 a.m. and 6 p.m. 1,866,387 pounds of water at 185 degrees into steam of 160 pounds gage. Eight men shoveled the coal to make them do it, and five men with two horses brought the coal into the boiler-room. If my figures are correct, this represents more than 5800 horse-power, or over 50 per cent. overload, and yet the flue gases went to the stack at 250 degrees only. What does Mr. Lane say about overload capacity?"

Twenty of these boilers set ready for steaming cost $38,600, which was much below the price at which any water-tube boilers were offered per unit of capacity. An analysis of the flue gases made on the day of the run above mentioned is given in the table below.

We are glad to see the matter discussed and hope that our contributors will make some figures, based upon the article, the additional data given by Mr. Diman and their own experience, and get them to us early so that they may be used in the May number. We hope also that Mr. Diman will have something further to say by that time.

RECORD OF ANALYSES OF THE FLUE GASES MADE MARCH 7, 1907, AT WOOD WORSTED MILLS, LAWRENCE, MASS.

Sample No.	Time.	Gases Entering Economizer. Deg. Fahr.	Gases Leaving Economizer. Deg. Fahr	Water Entering Economizer Deg. Fahr.	Water Leaving Economizer. Deg. Fahr.	Carbon Dioxide. CO_2	Oxygen. O_2	Carbon Monoxide. CO
A	10.05 a.m.	.....	.....	.....	.....	8.1%	11.0%	0.0%
B	10.35 "	460	250	105	185	6.0	13.7	0.1
C	11.05 "	458	250	110	183	5.4	14.2	0.0
D	2.25 p.m.	430	225	105	170	8.4	10.9	0.1
E	3.10 "	458	245	110	180	6.9	12.8	0.0

When sample D was taken only one fan was running; at other times both fans were in use. Sample E was taken just after all grates had been shaken. The samples were all taken from main flue just before entering the economizer.

Selecting an Indicator

Correspondents frequently ask us to select an indicator for them, or, rather, to advise them which of the several makes of indicators offered they would do best to buy. An instinctive realization of our position in the matter, not so well defined as to prevent them from writing, usually leads them to add that our advice will be considered strictly confidential.

Some engineers consider a particular indicator conspicuously superior, the next man will perhaps be equally as earnest a partizan for another. It is largely a matter of habit or bringing up, and of the purpose for which the instrument is desired. No indicator is good to buy if it will not do what the purchaser wants it to do. If his needs are simple, a simple and inexpensive instrument will fulfil them. A general idea of the working of the valves, of the steam distribution, and a fairly correct estimate of the power developed could be had from an instrument upon which less money and work had been expended to make it an instrument of precision than would be required upon an instrument which was to determine the acceptance or rejection of a large engine, or fix a bonus or premium or determine the data for a heat balance.

The thing for an intending purchaser to do, then, is to select an instrument of

the degree of precision required; test it for parallelism and freedom from friction; prove the correctness of the pencil movement, and either test himself or require from the maker test cards of the springs, if real accuracy is essential. If the instrument is fitted with a reducing motion, this should also be examined with especial reference to its ability to take up the cord as fast as it comes, without so much spring tension as will cause stretching and bothersome breaking of cords.

Some buyers will insist upon absolute simplicity, preferring, for instance, to depend upon their own dexterity in hitching and unhitching the cord to having the instrument encumbered with a detent motion. Others will want everything that ingenuity can apply, even to a rubber hose to carry off the blow-back.

Shakespere did with a quill, of which a penny would buy a handful, work which the world has been writing at ever since. He would have done no better with a gold-mounted Waterman. It is not the kind of an indicator which one buys which will determine the wisdom of his purchase, but the work that he does with it.

There is this to be said for the high-class instrument, that if one is competent to appreciate and care for it, it will do all classes of work, the precise as well as that of less importance; and it should also be said that the degree of refinement is not always in proportion to the price. Each instrument should be tested out and considered upon its merits.

The Pumps of the Two-stroke-cycle Engine

It is a well recognized fact that one of the drawbacks of the two-stroke cycle in power-plant work is the necessity for complicating the design and construction of the engine by the addition of pumps for delivering the gas and air to its power cylinder. In engines of considerable size this added mechanism is by no means an unimportant item. However, the separate pumps driven from the engine crank-shaft have the advantage that they may be of much less massive construction than the power cylinder and the allied parts. Consequently, the pumping is done by apparatus designed for that work only, instead of being done by machinery designed for the heavier work of converting heat energy into mechanical motion, as is the case with engines working on the four-stroke cycle.

In order to restrict the two-stroke-cycle engine to its principal functions as a prime mover and to reduce the weight and space per unit of useful output, it has been proposed to divorce the engine and its pumps, supplying air and gas to all the engines of a plant from a central pumping equipment, just as in a plant of steam engines running condensing the air and circulating pumps are sometimes centralized. This plan has been actually adopted by one of the American builders of steam engines who have recently entered the gas-engine field and another newcomer in that field is considering the plan. It obviously has the advantage of greatly reducing the total number of moving parts in a given plant as well as the floor space required, assuming that the plant comprises several engines. Moreover, a central pumping plant could supply compressed air for starting up the main engines, thereby eliminating the small air compressor now used and merging its air tank with the larger reservoir from which the engines would be supplied during normal operation. As the size of a gas engine has relatively small influence upon its efficiency, the thermal loss incurred by driving the pumps with an engine smaller than the main engines instead of driving direct from the latter would undoubtedly be more than compensated by the gain in mechanical friction due to the reduction in the number of pumps operated.

The proposition is an interesting one and opens up quite a line of speculative argument for and against it. It is not strictly analogous to the case of the condensing steam engine and its air and circulating pumps because it is of great advantage to be able to regulate the condenser pumps and their exhaust steam is useful, whereas the gas-engine pumps have a definite amount of work to do for each stroke of the main engine, regardless of external conditions, and the exhaust heat from a separate pumping engine cannot be utilized in a gas-power plant as usually built. The case is more nearly analogous to that of the alternating-current dynamo and its exciter. Notwithstanding the well known advantages of the "complete unit" system of equipping a power station, it has been found more expedient to put in a central exciting equipment than to provide each alternator with an exciter in stations of considerable magnitude.

Denatured Alcohol

The prospect of utilizing denatured alcohol economically as a fuel for internal-combustion engines has been brightened immensely by the passage of a bill by Congress last month amending the free-alcohol bill which went into effect on January 1. The original bill required the pure alcohol to be drawn off from the stills into barrels for transfer to denaturing warehouses, and made no provision whereby denatured alcohol could be produced practically on a modest scale. The amendment makes it permissible to transfer the pure spirit from distilleries to denaturing warehouses by means of either pipe lines or tank cars; it also provides that at distilleries producing alcohol for denaturation only, and having a capacity of not more than 100 proof gallons per day, cisterns or tanks may be used instead of bonded distillery warehouses, and the alcohol denatured without transferring it to a denaturing warehouse. Of course, the distillation, storage and denaturation must be done under government regulations, and these are now in course of formulation. Both of the amendatory provisions here mentioned will have the effect of reducing the cost of producing denatured alcohol, and the latter of the two will also make its production economically practicable by many farmers who would have found it impossible under the original law. After the amendment goes into effect, September 1, 1907, progressive agriculturists will find it feasible to drive most or all of their farming machinery by heat energy derived from their fermentable waste. Cornstalks, corncobs, potato peelings, tomato scraps—in fact, all vege table and fruit refuse—can be utilized to produce denatured alcohol. According to reports of some experiments made in the middle West, a ton of green corncobs yielded eleven gallons of alcohol, and a ton of green cornstalks six gallons. In general, the fermentable matter in vegetables and fruits will yield one-half its weight of alcohol, but the proportion of fermentable matter varies so widely that it is scarcely practical to attempt an estimate of the average quantity of alcohol obtainable from a given weight of mixed farm or domestic refuse. Turnips contain from 6 to 12 per cent. of fermentable matter; potatoes, from 15 to 26 per cent. Asuming an average of a trifle more than 10 per cent. for mixed refuse, a ton of it would yield one hundred pounds of alcohol, containing something like 1,200,000 B.t.u., which is the equivalent of 100 brake horse-power-hours from a well designed alcohol engine. It is too early to attempt any prediction as to the probable cost of distilling and denaturing the alcohol on a moderate scale, but there is, as yet, no reason to doubt that the production of this fuel from farm refuse will ultimately be worth while.

As to the application of denatured alcohol for power purposes: It has been demonstrated by experience in Germany and experiment in this country that the inferior heat value of alcohol as compared with petroleum distillates is largely offset by the higher efficiency with which it can be worked in internal-combustion engines, as compared with the other fuels. This higher efficiency is due to the greatly increased compression that is practicable, and to the fact that alcohol vapor and spray mix with air much more readily and effectively than is the case with hydrocarbons. Consequently, the fuel consumption in properly designed alcohol engines should not be as great as the performance of gasolene engines and the relative heat value of alcohol would indicate.

A New Poppet-valve Engine

In his "Reminiscences," recently published in POWER, Charles T. Porter registers a failure of the Porter-Allen engine to meet his expectations as to economy and lays that failure to excessive clearance.

A high-speed engine with clearance volumes not exceeding one per cent. of

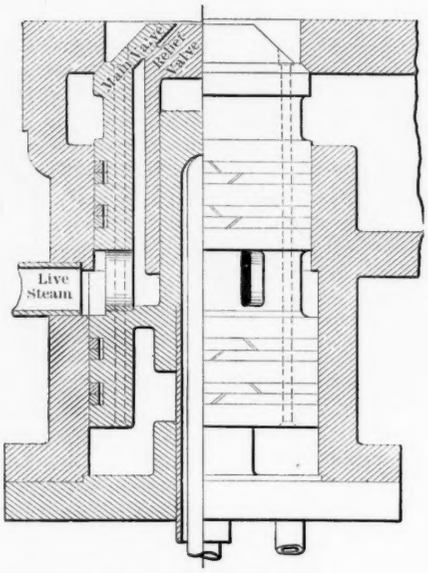


FIG. 2. EXHAUST VALVE

the piston displacement is what C. W. Crawford and James McBride have set out to accomplish by a renaissance of the poppet valve, the use of which although common in Europe has practically ceased in America, except for marine, pumping and other slow-moving engines.

The valves are seated in the cylinder wall as close as practicable to the bore, as shown by the section Fig. 1, and from

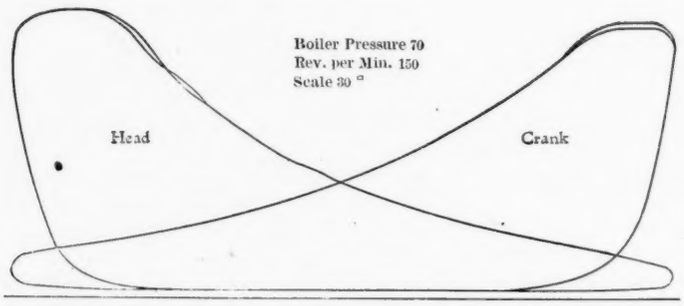


FIG. 3. DIAGRAM FROM CRAWFORD-M'BRIDE POPPET-VALVE ENGINE

the upper side trunks protrude through stuffing-boxes into the atmosphere. The valve is thus forced upward by the steam-chest pressure acting on the area of the trunk holding the inner end of the valve-operating lever firmly against the operating cam. The space above the valve piston is connected by a port with the space below the valve, so that however the pressure below the valve may vary, there is always available the steam-chest pres-

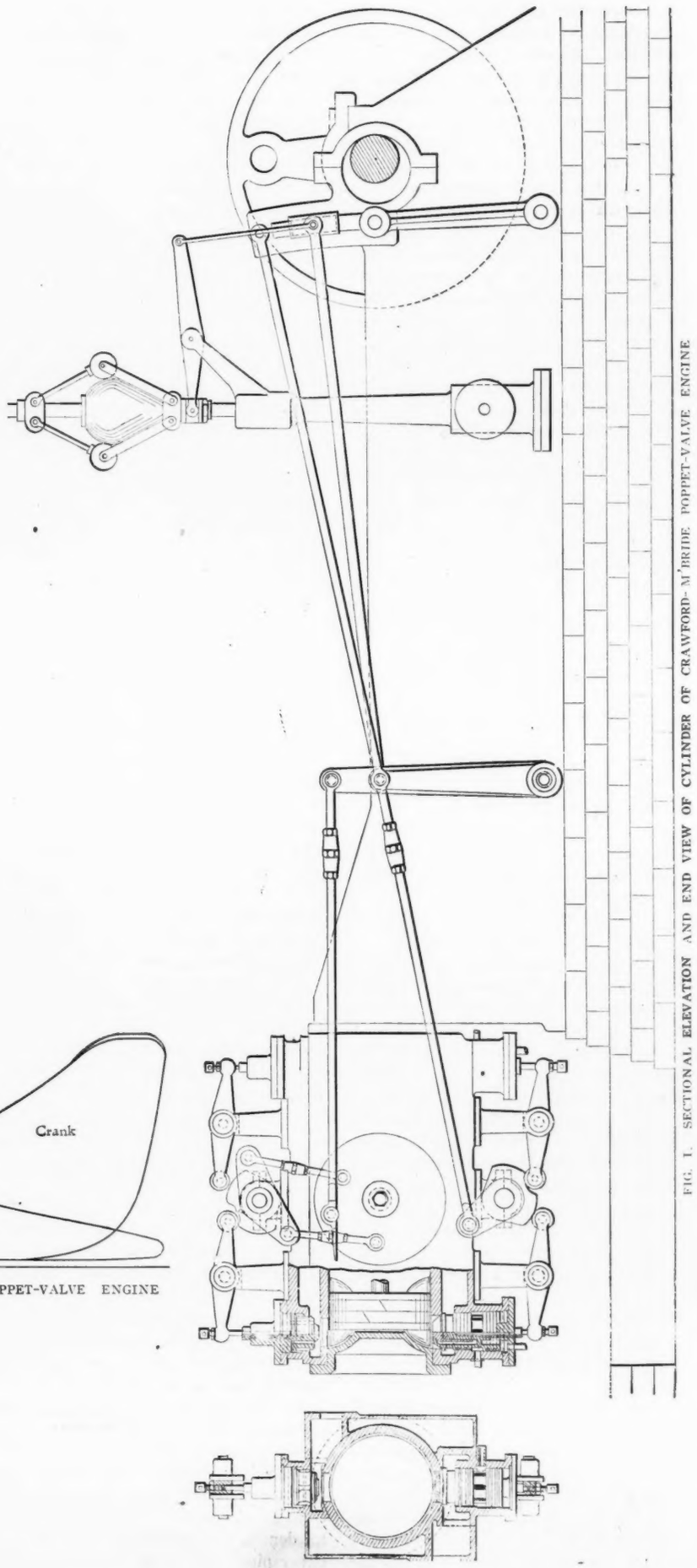
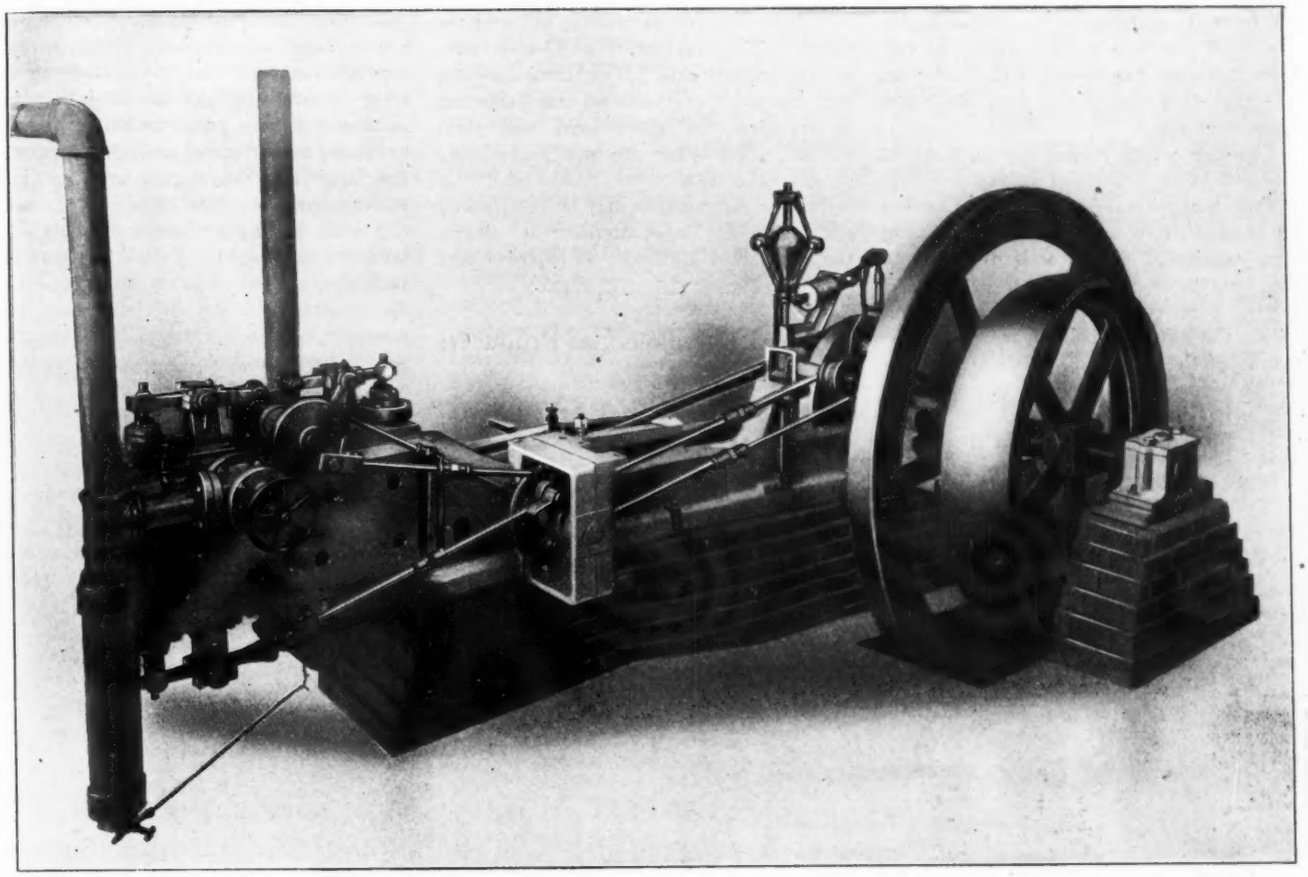
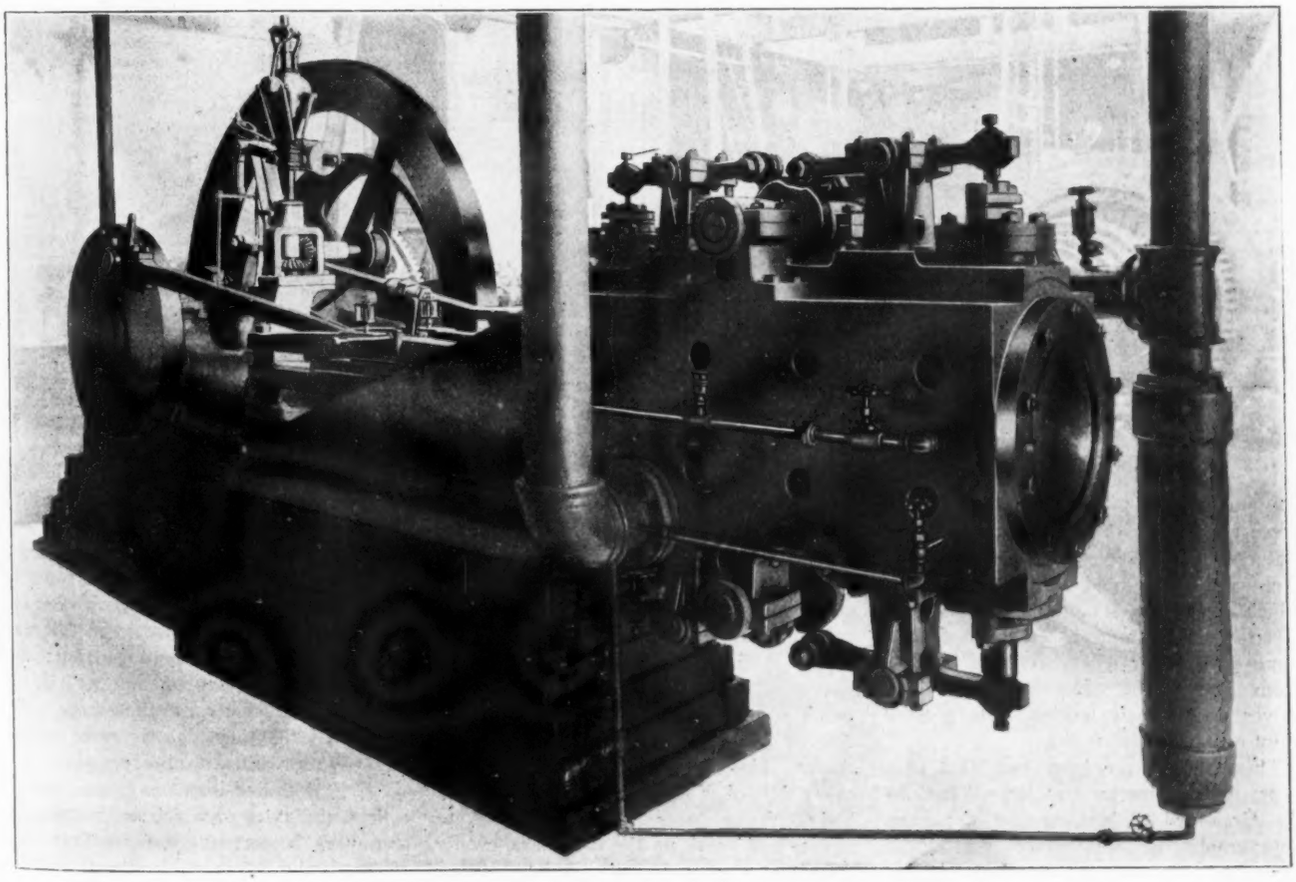


FIG. 1. SECTIONAL ELEVATION AND END VIEW OF CYLINDER OF CRAWFORD-M'BRIDE POPPET-VALVE ENGINE



THE CRAWFORD-M'BRIDE HIGH-SPEED POPPET-VALVE ENGINE



REAR VIEW OF CRAWFORD-M'BRIDE POPPET-VALVE ENGINE

sure on the area of the trunk to force it off from its seat, while the resistance of the small port to a sudden change of volume above the piston makes the latter act like that of a dash-pot to keep the valve from pounding.

The cam which forces the valve to its seat and holds it there at all times, except during the period of admission, derives its motion from a wrist-plate which, in turn, as shown in Fig. 1, is driven from the link block of a Porter-Allen governor, proportioning the time during which the valve is held open to the demands of the load.

The exhaust valve shown in section in Fig. 2 has live steam admitted to its interior, as shown, which steam forces the valve open, as does the steam chest the steam valves, and holds the relief valve closed. In the case of a charge of water the relief valve is pressed open and the water discharged through the pipe which admits live steam and which is best taken from a trapped separator on the same or a lower level. As will be seen from Fig. 1, the cams operating these valves derive their motion from a fixed point on the governor link.

These ideas have been embodied in a 14x30-inch engine built at Brazil, Ind., by the Crawford & McCrimmon Company, photographs of which are reproduced herewith. This engine has been run at 200 revolutions per minute, or one thousand feet piston speed and more, without jar, concussion, or perceptible noise of the valves. The diagram, Fig. 3, was taken at 150 revolutions. The cams, rollers, etc., are made of tool steel and hardened, as are also the valve-stems and their guides. The latter are not packed and do not need to be.

Further information may be obtained by addressing C. W. Crawford at Brazil, Ind., or James McBride, Stamford, Conn.

Hughes Continuous Gas Producers

We illustrate herewith three styles of the Hughes continuous gas producers, the product of the Wellman-Seaver-Morgan Company, Cleveland, Ohio.

Fig. 1 is a sectional view of the Hughes "mechanically poked" producer, the special feature of which is a mechanical poker composed of a water-cooled steel casting suspended from a trunnion, as shown in the illustration. This poker is operated by ratchet gearing actuated by a crank and crank-shaft driven by a reduction of gearing connected to the main shaft. This mechanism is designed to move the poker back and forth, while the body of the producer is being revolved, to agitate and break up the fuel mass, evenly distribute the coal, and aid in working down the ashes for removal. The main shaft and gearing are supported by a steel framework securely braced to the top of the producer. It has been found in practice, we are told, that the uniform treatment secured by the mechanical poker method not only produces better gas but a much higher rate of coal consumption per unit of gas area. Thus, under average conditions, the capacity of a hand-poked producer may be taken as ten pounds of coal per hour for each square foot of producer area, while the capacity of the mechanically-poked producer is said to average 25 pounds per square foot, and even 32 pounds has been maintained for long intervals.

The producer shell is of steel plate secured to a cast-iron base ring, to which is bolted a cast-iron water-seal, forming an ash receptacle. The base ring rests upon supporting columns bolted to a cast-iron revolving turn-table. This turn-table has at its outer circumference a cast-iron rack into which meshes a spur pinion keyed to a vertical shaft and connected to the main driving shaft by a train of gearing; thus the turn-table is rotated, and with it the body of the producer and the ash-pan. The bottom of the turn-table is fitted with a steel tread resting on the peripheries of six conical chilled-iron carrying wheels or rollers, the axles of which are mounted in stationary boxes. As the producer shell revolves, the ashes work down, and are deposited in the water-sealed ash-pan

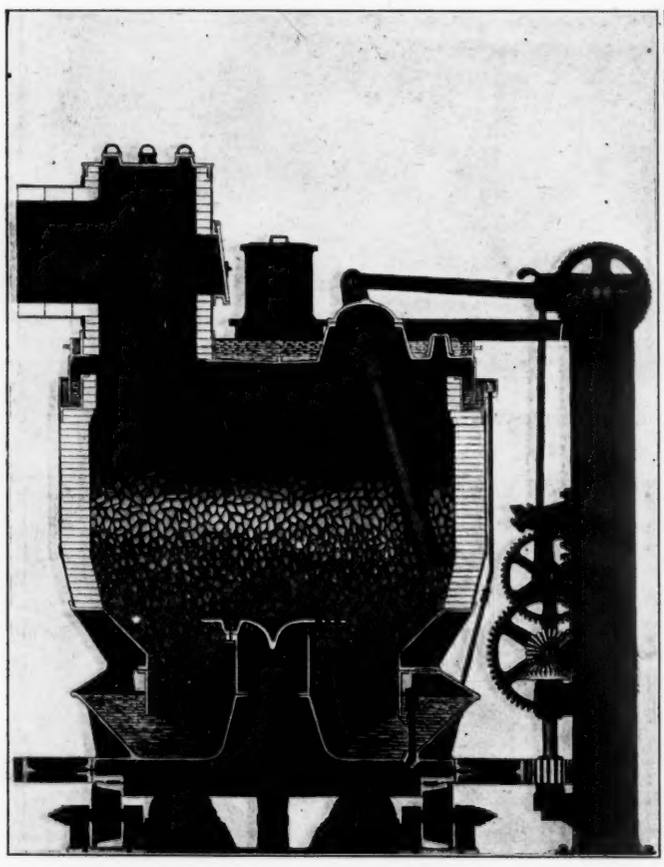


FIG. 1. HUGHES MECHANICALLY POKED CONTINUOUS GAS PRODUCER

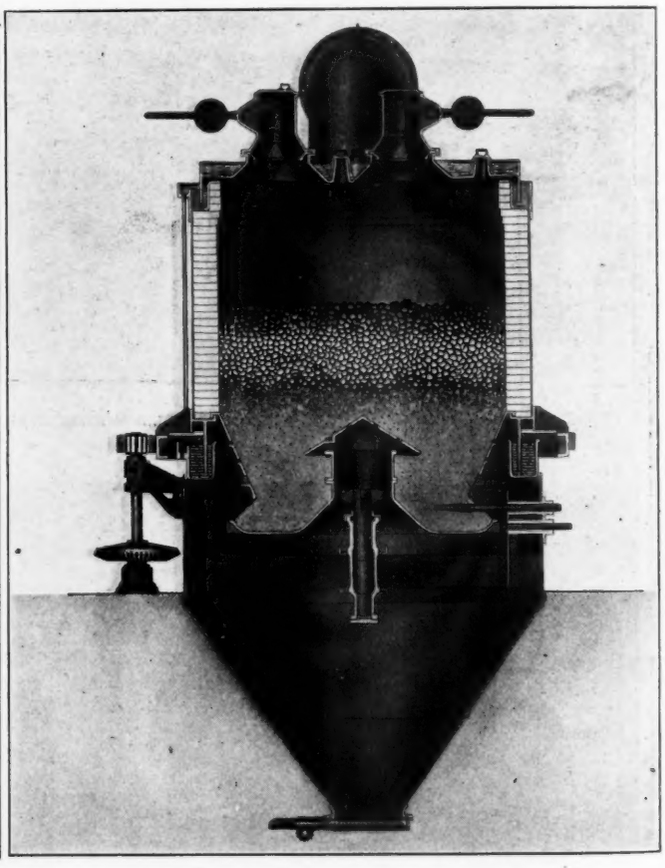


FIG. 2. HUGHES SELF-CLEANING CONTINUOUS GAS PRODUCER

from which they may be shoveled directly to the car. The producer is set upon and bolted to a concrete foundation. The producer shell is lined with fire-brick, unless otherwise desired by the purchaser.

The producer top is a steel casting flanged and ribbed to provide for water-cooling, a water-seal being formed by a top flange at the outer circumference of the producer cover.

Each producer is equipped with two feed hoppers, fitted with counterbalanced charging bells and gas-tight swing covers, placed at unequal distances from the vertical axis of the producer, so that the coal is deposited in concentric rings as the producer shell rotates, the object being to secure good distribution of the fuel. A gas outlet provided with a clearing door and peep-holes is attached to the top of the producer cover. This outlet has a short flanged neck to which connection from the outlet to the gas flue may be made. The base of the producer has a blast inlet pipe, as shown; a suitable blowing nozzle is provided, also a cast-iron deflecting plate for protecting the air opening.

These producers are driven by electric motors, or any other suitable power sources. The power required to operate a single producer continuously is said to be three electrical horse-power. For driving in groups a line shaft is recommended, from which each producer may be driven independently by means of a jaw clutch. While the actual labor required to operate a Hughes gas-producer plant depends on the arrangement and the facilities for coal and ash handling, it is stated that in large plants where the coal is delivered overhead into bins and fed to the producer by a hopper or other mechanical device, six men are sufficient to operate eight mechanically-poked producers. Tests of these producers, burning bituminous coal, are said to have shown a generally uniform quality of gas having from 3 to 4 per cent. of CO_2 and from 26 to 28 per cent. of CO. An average quality of gas can be maintained of the following composition by volume:

CO_2	CO	Hydrocarbons	H	N
4%	26%	3% to 4 %	13%	53%

Fig. 2 is a sectional view of a Hughes "self-cleaning" continuous gas producer, of the hand-poked type for anthracite coal; for bituminous coal the self-cleaning producer is equipped with the mechanical poker just described. The special feature of this type of producer is the method employed for mechanically disposing of the ashes and clinkers from under the water-seal. The producer is divided into two parts, upper and lower, with a water-seal between, the upper portion being rotated mechanically. By means of an iron bar thrust between the ash table and the revolving part of the body, the clinker is broken up and the pieces fall with the ashes into the cone-shaped ash receptacle beneath. By opening a large outlet valve at the bottom of this ash receptacle, the ashes are discharged into cars underneath.

The upper shell of the producer is made of steel plate riveted to and resting upon a steel ring forming a combined tread and rack. This rack is driven by a vertical shaft and pinion connected to the main driving shaft by a train of gearing. The thread of the rack travels on conical chilled-iron wheels having tight axles revolving in self-oiling boxes. As in the producer described above, the shell of the self-cleaning producer is lined with fire-brick, the top is a steel casting flanged and ribbed for water cooling, and two feed hoppers are provided and arranged as already mentioned. Also, the mechanical poking device, where employed, is similar to that previously described. In the hand-poked producer, however, the top is provided with a number of poke-holes

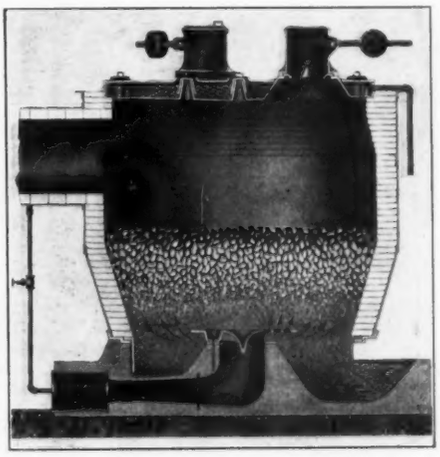


FIG. 3. HUGHES CONTINUOUS PRODUCER, HAND POKED

which are fitted with loose water-sealed covers, distributed so as to facilitate breaking up the surface of the mass of coal and keeping it properly agitated.

In this self-cleaning type the ash valve is operated by a rack and pinion controlled by a hand-wheel, and the ashes can be discharged without stopping the producer. Inside of the lower producer shell and near the top is a steel ash table; also a center blast-pipe supported by a vertical shaft. This shaft rotates on a step in an oil-cup bearing suspended from cross beams secured to the shell. When it is desired to grind the ashes and break up the clinkers to facilitate their down feed to the ash receptacle, the ash table is locked by means of a pin passing through the producer shell, and an iron bar is thrust into the ashes through the opening provided for that purpose, thus breaking up any clinker which may have lodged above the ash table. This ash receiver is provided with observation doors, making the interior accessible for breaking the clinker with bars, if required, or for other purposes. The producer is provided with a suitable blower nozzle as shown in Fig. 2.

The Hughes self-cleaning producers are regularly built in two sizes, viz., 8 feet 6 inches and 10 feet in diameter, respectively, these being the dimensions inside the fire-brick lining. The capacity of the hand-poked self-cleaning gas producer, burning anthracite, will range from 10 to 15 pounds of coal per square foot of burning area. When provided with the mechanical poking device for use with bituminous coal, the capacity is said to range from 25 to 32 pounds, as previously stated.

Fig. 3 is a sectional view of the regulation Hughes continuous gas producer, hand poked. The body or shell of this producer is of steel, stiffened at the top with an angle-iron flange and arranged for fire-brick lining. The bottom of the producer consists of a massive cast-iron base ring supported by three concrete piers so located as to permit the removal of ashes from any side of the base, the concrete foundation forming the ash receptacle. This concrete ash-pan is filled with water, and a cast-iron flange extends from the base ring down into the water, forming a water-seal in the ash-pan, enabling the continuous operation of the producer without shut-downs for the removal of ashes. There are no thin iron plates exposed to the corrosive action of the water.

The producer top is a water-cooled steel casting resting upon the brick lining of the producer. This top is subjected to great heat and experience has demonstrated that the use of cast-steel for such parts reduces the expense of maintenance. The 8-foot 6-inch producer has a single distributing hopper, located at the center of the producer top. The 10-foot producer has three feed hoppers so located as to facilitate uniform distribution of the fuel. These hoppers also have counterweighted bells and are covered with gas-tight swinging lids. The producer top is provided with poke-holes having loose water-sealed covers. The poke-holes are located near the outer circumference of the top to facilitate cleaning the walls of the producer. The blower is embedded in the concrete ash-pan, as shown in Fig. 3, and a cast-iron deflector plate covers the air opening at the center. Around the outer circumference of the top of the concrete foundation is a protecting angle. The producer is fitted with a short neck to which the outlet to the gas flue may be connected. There are a number of the Hughes continuous producers in operation and they are said to be giving satisfaction.

To blue iron or steel, mix one part clean sand with one part powdered charcoal; heat evenly in a pan until the piece, which has first received its finishing polish and been covered by the mixture, comes to the desired color. When cool, wipe dry with a cloth.

The Atlas Gasolene Engine

The accompanying engravings illustrate the salient features of the gasolene engines now built by the Atlas Engine Works, of Indianapolis, Ind. The engine is of compact design, with a substantial frame, as Fig. 1 shows. It is of the single-acting hit-and-miss class, working on the four-stroke cycle. The inlet valve is closed by a light spring and opened by atmospheric pressure, but it has features which differentiate it from the ordinary automatic inlet valve. It is relatively large in diameter and very light in weight; consequently it opens with a relatively small degree of vacuum or "suction" in the cylinder. During the period when the exhaust valve is open, the inlet valve is held firmly closed by an auxiliary rocker-arm actuated by the rocker-arm which opens the exhaust valve, so that there is no liability to fluttering of the valve on its seat while the burned gases are being expelled. This feature not only tends to prevent back-firing, but protects the inlet-valve seat against deposits of carbon.

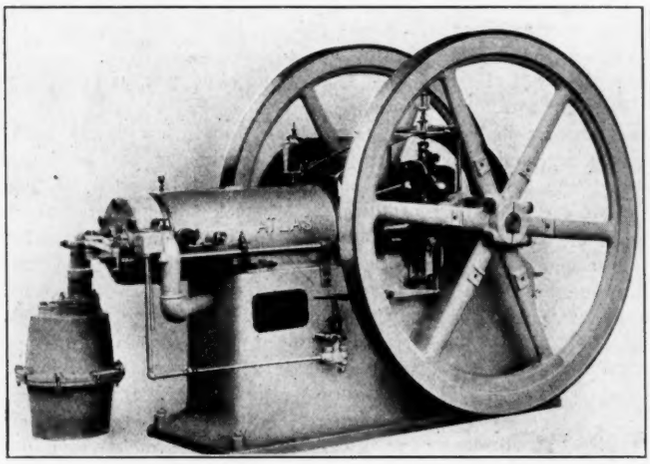


FIG. 1. ATLAS GASOLENE ENGINE

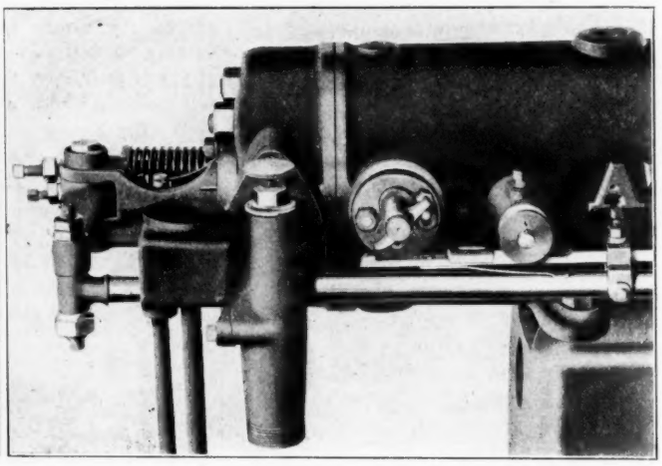


FIG. 3. SHOWING IGNITER HOOK AND TRIGGER

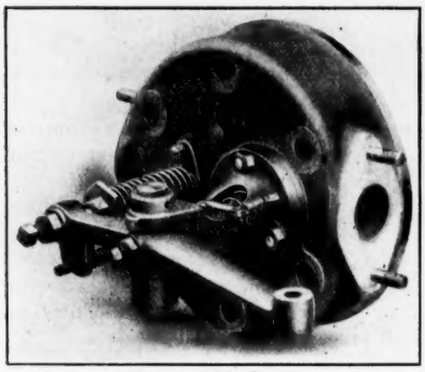


FIG. 2. CYLINDER HEAD

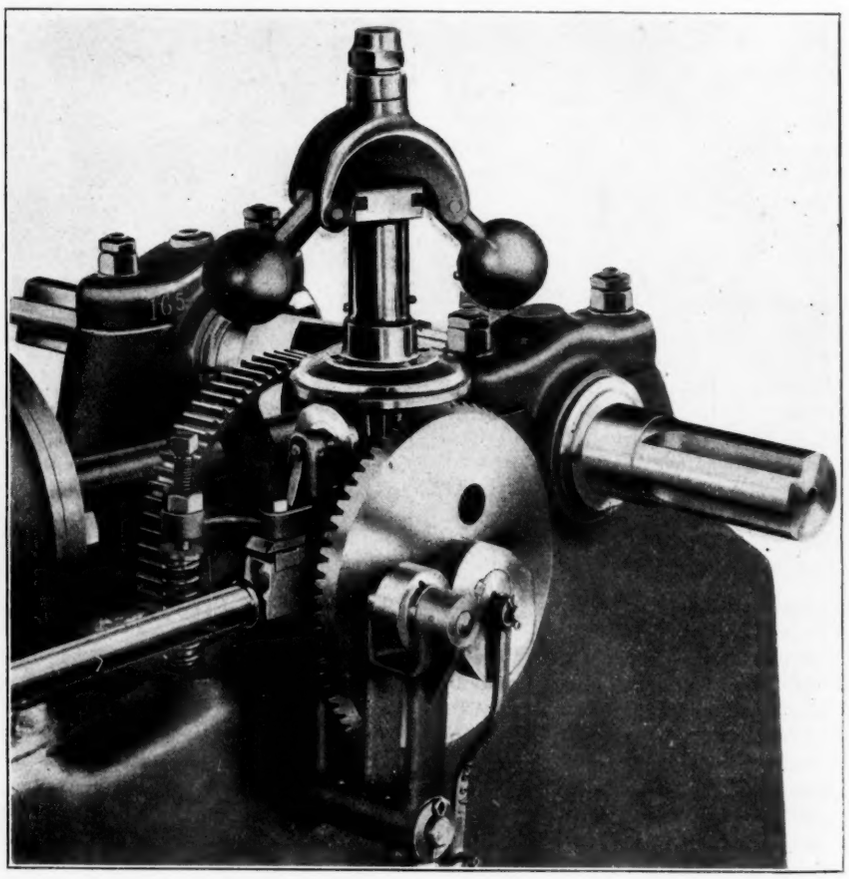


FIG. 4. GOVERNOR MECHANISM OF ATLAS GASOLENE ENGINE

The mechanism by means of which this locking is accomplished is shown in Figs. 2 and 5. The exhaust rocker-arm *A* is bifurcated at one end, one of the prongs serving to open the exhaust valve and the other to compress a strong helical spring, the other end of which rests against one end of an auxiliary rocker-arm *a*, pivoted at *p*. The other end of this latter lever presses against a thrust washer on the end of the inlet-valve stem and holds the valve tightly on its seat. When the bifurcated end of the exhaust-valve rocker-arm moves outward, allowing the exhaust valve to close, it relieves the pressure on the auxiliary rocker-arm *a* and allows the small spring *s* to pull the end of the arm away from the collar on the end of the inlet-valve stem, leaving the inlet valve under the control of its regular seating spring surrounding the stem, and therefore free to be opened by the suction of the piston.

The governor is of the fly-ball type, as shown in Figs. 1, 4 and 6, driven by bevel gears *g* and *G* from the secondary shaft, which is very short and mounted parallel to the crank-shaft. As long as the load is normal, the lever *L* is oscillated by the cam shown on the end of the shaft in Fig. 4. This lever moves the reach-rod *R* back and forth, operating the igniter and the exhaust valve. Should the speed exceed the normal, the governor balls will rise, depressing the sleeve, and with it the disk *d*, Fig. 6. This disk then de-

presses the frame *f* through the medium of the roller *r*, moving the latch *b* into the path of the dog *c* which is carried on the reach-rod *R*. The latch holds the reach-rod back and thereby holds the exhaust valve open and the igniter finger out of action; as the inlet valve is held closed while the exhaust valve is open, the cylinder can take in no charge, and the speed of course decreases until the governor releases the reach-rod and explosions are resumed.

The igniter is of the make-and-break type, the rocking electrode being actuated by a spring finger on the reach-rod, as clearly indicated in Figs. 1 and 3. This finger runs under a roller mounted eccentrically on a stationary stud and by changing the position of this roller on its stud the point of ignition can be adjusted.

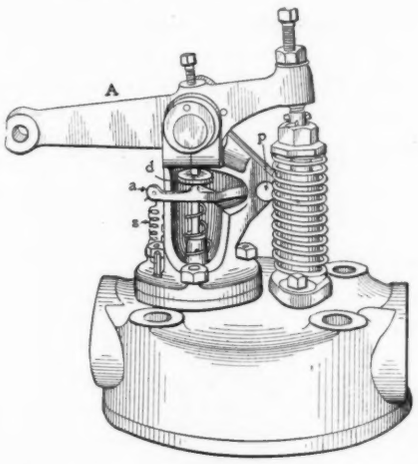


FIG. 5. VALVE MECHANISM

A special vaporizer is used in which there are no moving parts. Gasolene is supplied to the vaporizer by a small pump (shown near the fly-wheel in Fig. 1), and the vaporizer is provided with an overflow outlet the level of which is a trifle below the level of the nozzle from which gasolene is sprayed into the inrushing air by the suction of the latter. This arrangement obviously gives a constant level in the reservoir of the vaporizer and prevents flooding at light loads.

The piston is ground to size and provided with four packing rings of the usual spring type with overlapping ends at the cut. The gudgeon-pin is secured in the piston, and the connecting-rod is provided with an adjustable bronze-bushed box, as indicated in Fig. 7. This engraving also shows the oil wiper attached to the piston end of the rod; an oil tube, mounted inside the front end of the piston, takes oil from a cup mounted on the cylinder and delivers it to the wiper. The oil cup referred to also lubricates the piston.

All of the bearings are bushed with a special bronze and automatic lubrication is, of course, provided throughout.

The method of securing the fly-wheel to the shaft is unusual. In order to avoid the "chewing" effect entailed by the ordinary method of keying unless the key is an absolute fit in both keyways, two keys are used and the keyways are cut as shown in Fig. 4. It is claimed for this arrangement of keyways that the stress on the keys is one of compression alone as distinguished from the shearing stress on a single key located in the ordinary manner. This is due to the fact that the seams along the joint between the keyway in the wheel hub and that in the shaft coincide with the corners of the key, instead of lying along its plane surfaces. The fly-wheel has a split hub, so that the keys can be tightened when necessary.

The crank-shaft and crank are made from a steel bloom in one piece, the loop formed by the crank-pin and cheeks being cut out of the solid instead of being bent to shape.

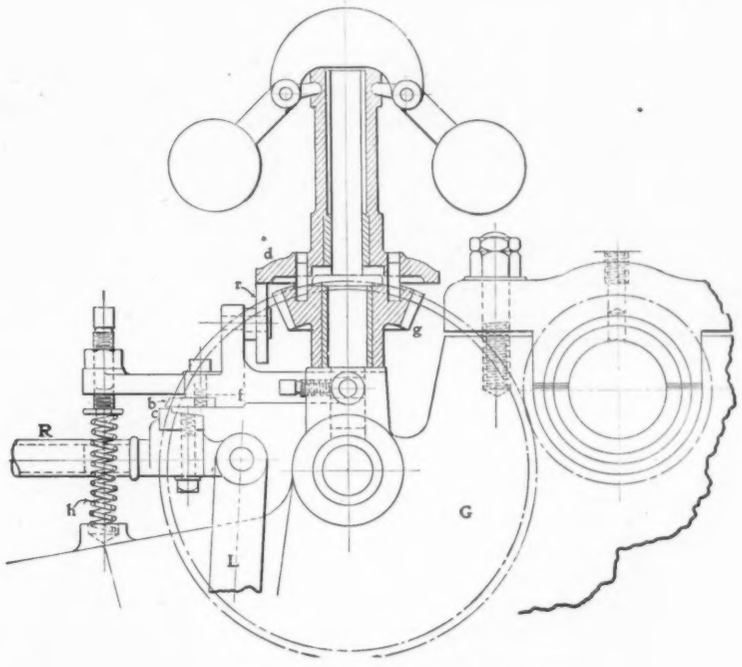


FIG. 6. GOVERNOR DETAILS, ATLAS ENGINE

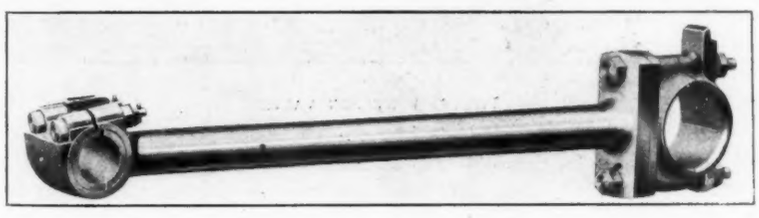
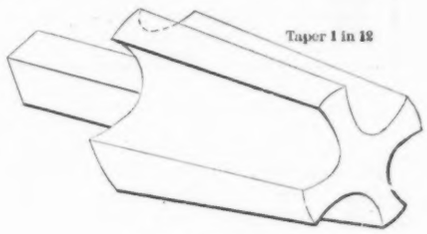


FIG. 7. ATLAS CONNECTING-ROD

Boiler-tube Plugs

A correspondent of the *American Machinist*, commenting upon Chas. T. Porter's account in his "Engineering Reminiscences" of experiments made by a practical engineer to determine the distribution of heat passing through the tubes of a horizontal return-tubular boiler, says that the effect noted by Mr. Porter is verified by his own experience, and describes how he increased the efficiency of his boiler by the use of retarding plugs, made of cast-iron and fashioned as shown in the accompanying engraving. These were inserted at the front end in the tubes which appeared to carry an excess of the gas, so that the gas was forced around through the other tubes and more uniformly distributed over the heating surface.



BOILER-TUBE PLUG

The *Engineering Journal* says: The power efficiency of certain soft coals when used in the gas-producer plant is two and one-half times greater than when used in an ordinary steam-boiler plant.

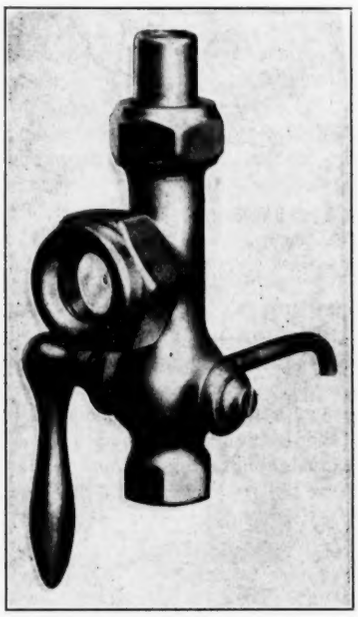
Power Plant Machinery and Appliances

Original Descriptions of Power Devices
No Manufacturers' Cuts or Write-ups Used

MUST BE NEW OR INTERESTING

The "Multiplus" Sight Feed

The "Multiplus" sight feed, which has just been put on the market, is a combination sight feed and three-way cock, a design calculated to enable the engineer to see at a glance whether oil is being pumped to the cylinder or point of delivery, and by simply turning the handle to determine the rate at which oil is being fed. This device is simple in construction; on one side of the main barrel is a circular chamber, the flat wall of which is a heavy glass disk against which presses a hollow plunger. In its passage from the lubricator to the point of delivery the oil is forced through this plunger, which has



THE MULTIPLUS SIGHT FEED

an opening in the end that presses against the glass. The oil in forcing its way between the glass and the end of the plunger moves the plunger back so that only the oil is seen behind the glass, but as soon as the pressure is relieved the plunger resumes its normal position, and the light color of the brass is seen instead of the dark oil. Thus, when the oil is being fed, the sight feed will keep up a continual "winking." There is always a solid column of oil under pressure.

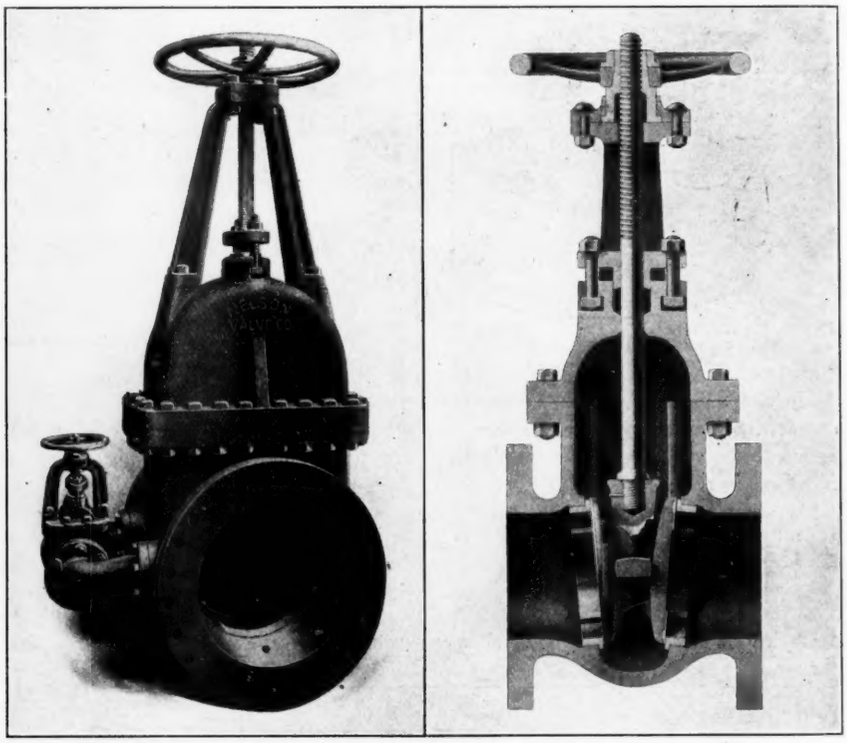
The "Multiplus" sight feed is made by Greene, Tweed & Co., 109 Duane street, New York, and is furnished with their improved "Rochester" automatic lubricators.

The New Nelson Valve

The principal feature of the Nelson valve is the freedom of the disks to move into any required position to adapt themselves to their seats. The valve body is provided with converging bronze seats screwed in before facing and fitted with two bronze-faced disks entirely independent of each other, and free to move in any direction. These are suspended upon a central wedge, which being forced between them by pressure from the stem, compels a tight joint regardless of distortion from unequal expansion or any other cause. This is the distinguishing feature of the "Nelson valve," as it has been known upon the market for some years.

As shown by the accompanying engravings the valve has undergone a complete revision in its general design, its previous rectangular lines having been changed to the elliptical, resulting in uniform section and freedom from foundry strains, as well as a more pleasing valve to look at and a more comfortable one to work around. The elliptical form of the bonnet makes possible, too, the interposition of a rail engaging with lugs upon the backs of the disks, holding them firmly in place and preventing clattering when the valve is open.

The position of these rails and the method of their operation will be apparent from the section. The disks bear on the wedge in a single line, can readily swing into the plains of their respective seats or accommodate themselves to the presence of grit or foreign matter, and should one disk be prevented from seating from this cause the other will make a joint and close the valve. The angle of each seat is five degrees from the vertical. The arrangement of the disks, as described, demonstrates the advantages of this valve when used as a blow-off valve. When desired, nickel-steel stems and spe-



THE NEW NELSON VALVE

cial nickel-composition seats are used. No stud bolts which are liable to twist off and have to be drilled out are used, all the attachments being by bolts in slots as shown in the engravings, and should one of these fail, a common bolt out of stock will replace it. The thread has been changed from square to the Acme 29 degree. Each valve is looked upon by the manufacturer as an individual machine, thoroughly tested and given a consecutive number, of which a record is kept so that the valve or any of its parts can be promptly and accurately duplicated by reference to the number.

The Nelson Valve Company, Wyndmoor, Philadelphia, Penn., is the manufacturer.

Allis-Chalmers Oil-insulated Transformers

Two types of the line of power transformers built by the Allis-Chalmers Company, Milwaukee, together with some of the principal details of each, are illustrated by the accompanying engravings. Fig. 1 is a sectional elevation of the self-cooling oil-filled type, which is built in sizes up to 300 kilowatts. Fig. 2 shows the core and coils assembled ready for mounting in the case, and Fig. 3 illustrates the method of setting the coils into the core base or stool. From these engravings it is obvious that the transformer is of the shell type. The coils, both primary and secondary, are of the "pancake" type, as illustrated by Fig. 4, which shows two of the low-voltage coils, one before and the other after the final taping. The primary and secondary coils

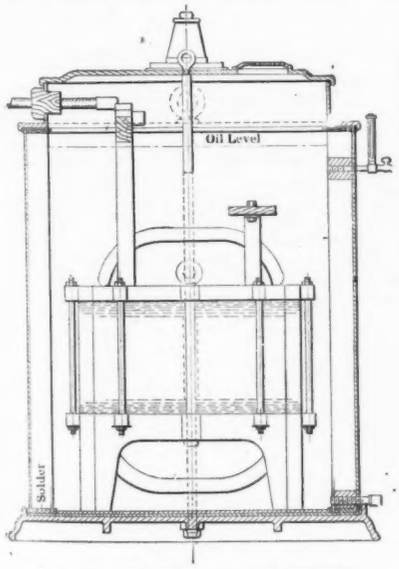


FIG. 1. OIL-COOLED TRANSFORMER

are sandwiched, of course, in order to minimize magnetic leakage and to present the maximum surface to the oil for cooling. The core is built up with sheets 0.014 inch thick, each of which is varnished to prevent eddy currents in the core.

The electrical and magnetic construction of the oil-filled, water-cooled type of transformer is practically identical with that of the oil-cooled type just described. The difference between the two consists in the addition of a coil of seamless brass or copper tubing which surrounds the core and coils of the transformer, and through which water is circulated to carry away the excess heat. Fig. 5 is a sectional elevation of a complete transformer of this type. Both ends of the pipe coil are brought out through stuffing-boxes in the case at points below the oil level; this is done to avoid the liability to freezing and bursting while a transformer is not in service. The pipe coil is of such diameter that the transformer core and coils can be lifted out of the case without disturbing the pipe.

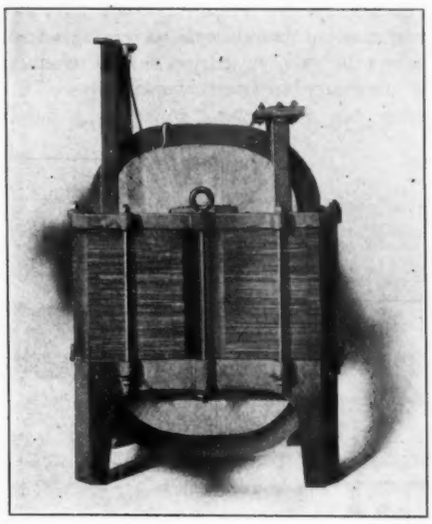


FIG. 2. TRANSFORMER CORE AND COILS

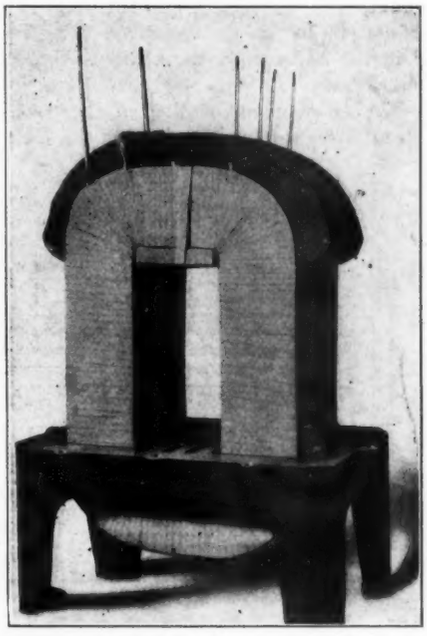


FIG. 3. COILS READY FOR CORE ASSEMBLY

New Automatic Gage-glass Fitting

Last year there was patented—and it has just been placed on the market—a combination water gage and automatic gage-glass safety valve designed chiefly to prevent personal injury and property damage resulting from exploded or otherwise broken gage glasses. The invention combines a water gage and two automatic safety valves, each with a single casing, which being directly connected to the boiler or water column serve to eliminate the old-time globe valve of the water gage, as well as other connecting parts.

FIG. 4. TWO LOW-VOLTAGE COILS

A prominent feature of the appliance, emphasized by the makers, consists in the valves being so constructed that it is impossible for an operator to close them by external means, either through ignorance or carelessness. When, however, the equilibrium of pressure in the gage glass is destroyed, as by the glass breaking, the valves are instantly closed by the steam

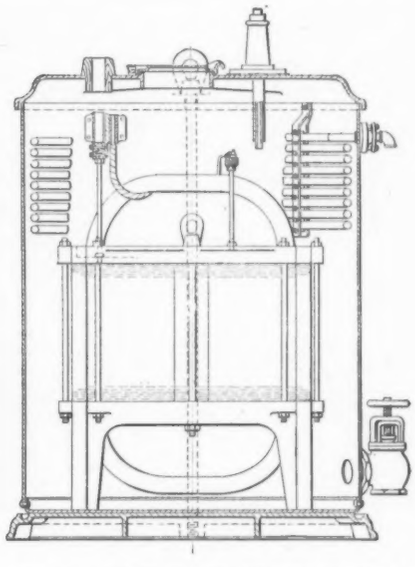


FIG. 5. WATER-COOLED OIL-FILLED TYPE

and water pressure behind them. It is stated that the action is so spontaneous, moreover, that the broken glass does not have a chance to "fly," even, and the danger of steam escaping is wholly obviated. When a new glass is put in, which may be safely done at the engineer's convenience, the valves automatically open as soon as the boiler pressure again becomes equalized.

It is also pointed out that while the valves cannot be forced shut, they may be forced or held open when blowing down the gage. Otherwise it is seldom necessary to force the valves open, for the reason that the disk of each valve is provided with a port, so that it will admit the passage of a small quantity of steam and water into the new gage glass, causing the glass to fill within five minutes after the pressure balance has been restored. Then, there being nothing to hold the valve closed when the glass is full, it opens by gravity.

A still later improvement provides a small port in the valve hinge, so that any sediment that may settle below or in front of the valve seat will be forced out when the water gage is blown down. The port between the boiler and gage glass is about four times as large as in an ordinary water gage.

Fig. 1 is a sectional view of a ¾-inch lower valve of the combination set, showing the valve disk in normal position. The disk swings on a hinge pin, expanded on which is a shoulder that abuts a like shoulder on the valve hinge. The enforced action of these shoulders on each other withdraws the valve disk from its seat when it is necessary to open it manually. The hinge pin is operated either by a hand wheel or by a bar and chain (see Fig. 2) on the outside of the casing, the latter attachment being necessary where the water gage is out of the reach of the engineer. There is a bar on the top valve, also. To hold the valve open the chain is simply pulled down.

The opening and closing of the cocks operates the safety valves, and at the same time tests them as to their working condition. Also by opening the cock in the lower valve cap, a direct circulation is obtained through the lower valve. The valves are tested to withstand a hydrostatic pressure of 700 pounds, which is equivalent to 300 pounds steam pressure. They are made in all sizes.

This combination water gage and automatic fitting is the product of the Northern Specialty Company, 63-65 Columbia street, Utica, N. Y.

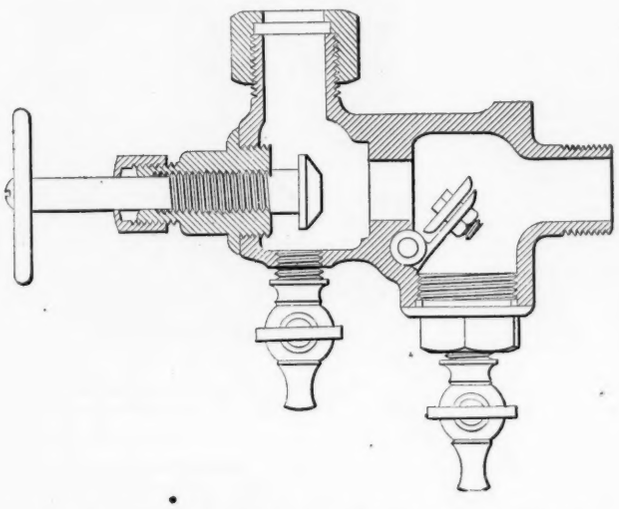


FIG. 1. SECTIONAL VIEW OF LOWER VALVE OF AUTOMATIC GAGE-GLASS FITTING

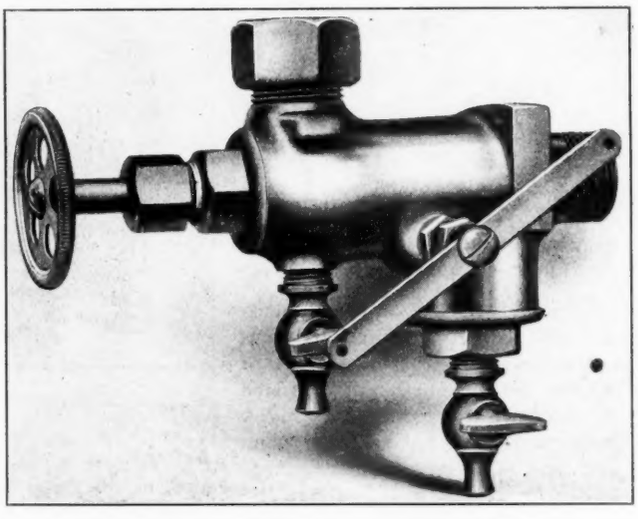


FIG. 2. OUTSIDE VIEW OF LOWER VALVE OF AUTOMATIC GAGE-GLASS FITTING

Robertson's Low-headroom Boiler

In designing the type of boiler herewith illustrated, the object of the inventor, W. Robertson, of Milton, Mass., is to provide a boiler suited for use in low basements and other places where insufficient headroom prohibits the installation of the water-tube or safety type of boiler, and where provision is essential for greater safety than is afforded by boilers with shells of large diameter. The invention embodies, in effect, a combination tubular and sectional boiler, as being deemed most practicable for carrying steam at high pressure with safety, while securing maximum capacity for the space occupied. Fig. 1 shows a vertical cross-section of the boiler and furnace through

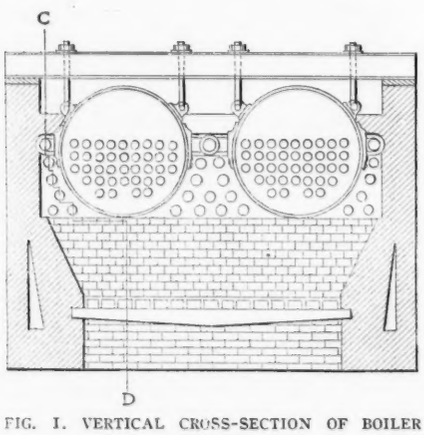


FIG. 1. VERTICAL CROSS-SECTION OF BOILER AND FURNACE

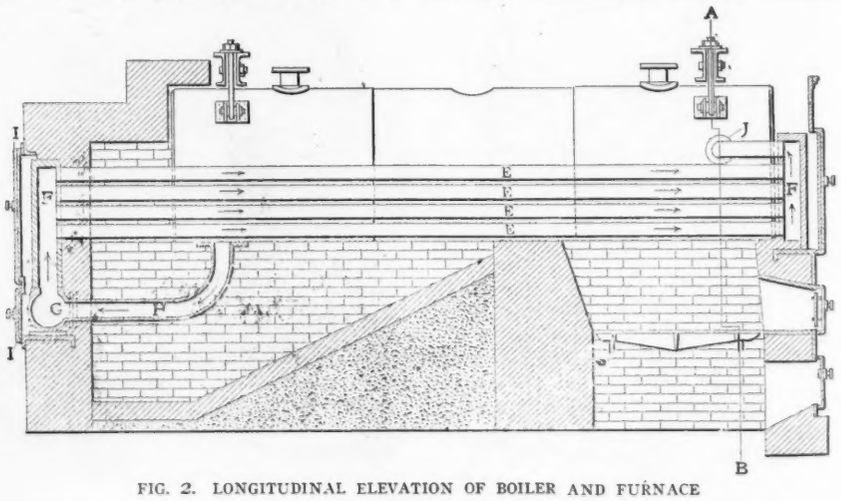


FIG. 2. LONGITUDINAL ELEVATION OF BOILER AND FURNACE

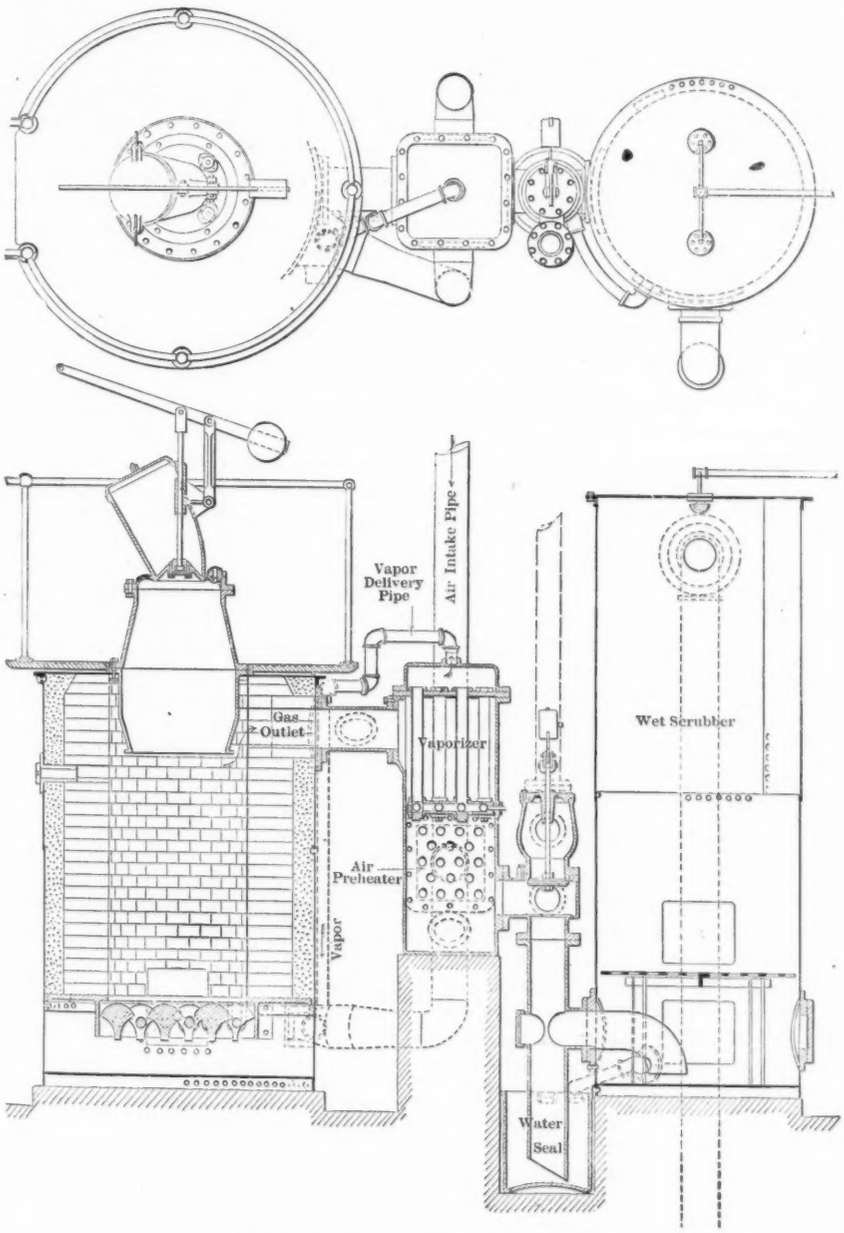
the line *A—B* on Fig. 2; the latter is a longitudinal elevation (side wall of the setting removed), substantially through *C—D*, Fig. 1.

The construction includes either one or two shells of small diameter provided with fire-tubes, as shown in Fig. 1. The shells are suspended from overhead beams, and at each side of and between the shells is incorporated a system of horizontal water tubes *E*, Fig. 2. The horizontal water tubes are expanded into headers *F* at the front and rear ends of the boiler (the headers have been drawn for illustrative purposes only), the headers at the rear end being expanded into a mud-drum *G*. The mud-drum is connected by large pipes *H* riveted to the bottom of the rear end of the shells. The front headers may be constructed as manifolds, riveted to the shells, and in alinement with the front boiler. *I, I* represent metal doors lined with non-conducting material.

It is expected that with this construction a continuous flow of water will be promoted from the shells through the large pipes *H* into the mud-drum *G* and the rear headers *F*, thence through the water tubes *E* into the front headers and into the shells at *J*, as indicated by the arrows in Fig. 2.

The Munzel Suction Gas Producer

The Minneapolis Steel and Machinery Company, Minneapolis, Minn., is building a very simple line of suction power gas producers designed by Max Munzel, of Braunschweig, Germany. The accompanying engraving illustrates the constructional features of the producer. The generator is a simple cylinder lined with fire-brick and provided at the bottom of the fuel chamber with rocking grate-bars somewhat similar to a familiar steam-boiler grate. From the upper part of the generator, the gas passes to a vaporizer consisting of a rectangular box containing a nest of vertical tubes which open into a vapor chamber at the top. The tubes contain the water, which the hot gases heat and evaporate on their way through the box. In the lower extension of the same box is the air preheater consisting of a second nest of tubes, arranged horizontally and opening at each end into side chambers; air enters one of these chambers and passes through the tubes to the other chamber, and thence to the ash-pit. The pipe which takes steam from the upper chamber of the vaporizer delivers it to the air flue just outside the generator casing and the mixed hot air and steam pass to the ash-pit.



THE MUNZEL SUCTION GAS PRODUCER

The gases after passing through the vaporizer and preheater are led to a wet scrubber in the usual way, and the usual atmospheric vent valve is provided in the pipe connecting the preheater and scrubber. Below the vent valve is a settling tank which water-seals the outlet for such impurities as are not carried to the scrubber by the current of gas. The scrubber is of the ordinary coke-filled type, supplied with a water spray. Beyond the scrubber is an expansion tank (not shown here) through which the gas passes to the engine, the function of this being to smooth out the pulsations due to the intermittent drafts by the engine piston by taking in the current of gas started by the engine suction during the brief period while the admission valves of the engine are closed. The slight reserve capacity of the tank also reduces the violence of each suction effort by the engine piston so far as the generator is concerned, thereby tending to smooth out the pulsations still further.

To Our Canadian Subscribers

The present postal treaty between Canada and the United States expires on May 7 next. The Canadian Government has before it a proposition to increase the rate of postage on periodicals from the United States to such an extent as will make it necessary for us to increase our subscription rate to Canada to $3.

If Canadian friends of this paper object to such a raise it would be well if they wrote their postal authorities protesting against such action.

Much annoyance was recently caused at a railway plant from excessive priming of the boilers and marked signs of corrosion. Upon investigation these were found to be occasioned by large quantities of fat which were dumped into the brook from which the feed-water was taken by a tannery a couple of miles up stream.

Inquiries

Questions are not answered unless they are of general interest and are accompanied by the name and address of the inquirer.

Allowable Pressure on a Boiler Flue

What is the proper working pressure on a flue 25 inches in diameter and 3.6 inches long, of ⅜-inch plate? J. E. L.

The English board of trade rule for the working pressure of circular furnaces with longitudinal joints welded or made with a butt strap is:

$$\frac{90{,}000 \times \text{square of thickness in inches}}{1 + \text{length in feet} \times \text{diameter in inches}},$$

provided this does not exceed:

$$\frac{8000 \times \text{thickness in inches}}{\text{diameter in inches}},$$

Substituting your figures:

$$\frac{90.000 \times 0.375^2}{4.5 \times 25} = 112.5 \text{ pounds},$$

$$\frac{8000 \times 3}{25 \times 8} = 120 \text{ pounds}.$$

Since the 112.5 pounds does not exceed the value given by the other formula, it is the working pressure allowed.

Speed of Transmission Rope

How is the speed of transmission rope calculated? It is a multiple drive, with nine ropes, 60-inch driver sheave running at 130 revolutions per minute; 30-inch driven sheave; 21 foot centers; tightener 40 inches; distance from driver to tightener centers, 16 feet; one-inch rope. G. A. U.

The circumference of a circle is 3.1416 times its diameter. The circumference of your 60-inch or 5-foot sheave is:

$$5 \times 3.1416 = 15.708 \text{ feet}$$

and as it makes 130 revolutions per minute the rope speed is:

$$130 \times 15.708 = 2042.04 \text{ feet per minute.}$$

The diameter should be measured on the pitch line of the sheave, i.e., on the line where the rope bears, where the rope is going as fast as the line in question.

A "multiple" drive is one in which a number of separate ropes are used, not one with a single rope passing a number of times around the sheaves. The multiple drive does not require a tightener, or if it did it would need one for each rope.

Safe Working Pressure for a Boiler

Please advise me as to the safe working pressure of a new boiler. The boiler is of the horizontal tubular type, 15¼ feet in length, 48 inches in diameter and constructed of steel 9/32 inch mean thickness, stamped Carnegie flange 60,000 T. S., with heads of steel 7/16 inch mean thickness stamped Carnegie flange 60,000 T. S. The longitudinal seams are lap double-riveted rivets made of steel; diameter of rivet holes ¾ inch; pitch of rivets 2.68; braces front head above tubes 5 size 1 inch, back head above tubes 4; below tubes 4 size 1 inch, style of braces crowfoot; number of tubes 36; diameter of tubes 3½; round-about seams are lap single riveted; pitch of rivets 2 inches. W. P.

The weakest place is liable to be the longitudinal seam. The plate efficiency of this seam is found by dividing the difference between the pitch and diameter of the rivets by the pitch,* in this case:

$$\frac{2.68 - 0.75}{2.68} = 0.72$$

or 72 per cent., and that of the rivets:

$$\frac{0.854 \times \text{diameter}^2 \times \text{shearing strength} \times \text{rim}}{\text{pitch} \times \text{thickness} \times \text{tensile strength}}.$$

You do not give any shearing strength. It is usually taken at about two-thirds of the tensile strength. Make it 0.64 of the tensile strength of the plate and the formula becomes:

$$\frac{0.5 \text{ diameter}^2 \times \text{rows}}{\text{pitch} \times \text{thickness}};$$

substituting the figures of your case:

$$\frac{0.5 \times 0.75 \times 0.75 \times 2}{2.68 \times 7/16} = 47.97.$$

This efficiency is very low for a double-riveted joint. The rivets are too small or the pitch too great for the thickness of the plate.

To find the breaking strength of the shell:

$$\frac{\text{thickness} \times \text{tensile strength} \times \text{joint efficiency}}{\text{radius}};$$

$$\frac{7/16 \times 60{,}000 \times 0.48}{24} = 525 \text{ pounds},$$

which, to give the allowable working pressure, should be divided by a factor of safety of from 3.5 to 5.

We cannot determine the adequacy of the bracing without knowing the distance between the top row of tubes and the top of the boiler. You will find the process explained in a "Simple Talk" upon the subject in our issue of October, 1905, on page 612.

Relation between Meter Readings and Engine Power.

In an electric power station the average reading of the voltmeter is 541 volts, and that of the ammeter is 86 amperes; what is the horse-power developed? If the efficiency of the generator is 92 per cent., and if 17½ per cent. of the indicated horse-power of the engine is wasted in overcoming frictional resistances, what must be the indicated horse-power of the engine? R. W. C.

Horse-power is equivalent to watts divided by 746; hence the electrical horse-power developed in your case is

$$\frac{541 \times 86}{746} = 62.367 \text{ horse-power.}$$

If 17½ per cent. of the engine's indicated horse-power is lost the efficiency will be 82.5 per cent.; the efficiency of the generator being 92 per cent., the combined efficiency of the two will be $82.5 \times 92 = 75.9$ per cent. Hence, 62.367 horse-power is 75.9 per cent. of the indicated horse-power, which must therefore be

$$\frac{62.367 \times 100}{75.9} = 82.17 \text{ horse-power.}$$

Effect of Load on the Speed of a Corliss Engine

Please inform me whether a Corliss engine runs faster under full load or no load, or is the speed the same? G. A. U.

A Corliss engine runs slower with the load on. If it did not the balls would not run in the lower plane necessary for the later cut-off.

Average and Maximum Values of Alternating Current

What is the relation between the average and the maximum values of an alternating current or e.m.f.? A. C. B.

The average value is 0.6366 of the maximum value, but it is of no practical importance. The working value is the *effective* current or e.m.f.; this is 0.707 of the maximum.

Utilizing Alternators of Different Frequencies

We have a 60-kilowatt alternator of 133 cycles and a 37-kilowatt alternator of 140 cycles. Can the two be operated in parallel or otherwise in conjunction? O. W. K.

It may be practicable to operate them in parallel by reducing the speed of the 140-cycle machine to 95 per cent. of its present speed, but it is probable that their characteristics are so different as to make satisfactory paralleling impracticable. There is no other way to operate them together with any degree of satisfaction.

Quantity of Water Evaporated by Heat from Burning Hydrogen

How many pounds of water, at 67 degrees Fahrenheit, can be changed to steam at a pressure of 90 pounds per square inch gage, by burning 7 pounds of hydrogen, assuming that 23 per cent. of the heat of combustion is lost? R. W. C.

Seven pounds of hydrogen when burned will liberate

$$7 \times 62{,}032 = 434{,}224 \text{ B.t.u.}$$

theoretically available for heating the water, and if there is a loss of 23 per cent., the net heat available will be

$$434{,}224 \times 0.77 = 334{,}352 \text{ B.t.u.}$$

The heat required to convert water of 67 degrees temperature into steam at 90 pounds pressure is

$$1182.84 - 35.02 = 1147.82 \text{ B.t.u.},$$

so that the quantity of water evaporated by the burning hydrogen will be

$$\frac{334{,}352}{1147\ 82} = 291.29 \text{ pounds.}$$

*See page 560, September, 1905.

Book Reviews

INJECTORS: THEIR THEORY, CONSTRUCTION, AND WORKING. By W. W. F. Pullen. Third Edition. 1906. The Technical Publishing Company, Limited, London and Manchester; D. Van Nostrand Company, New York. Cloth; 208+VI pages, 7x4¾ inches; 148 illustrations. Price, $1.50.

This valuable little book is too well known to require an extensive review; nevertheless, it being the most important work on the subject, the publication of a third edition deserves some notice. The author states, in his last preface, that a great deal of experimental work has been done on the steam jet, principally in connection with steam turbines, since the appearance of the first edition, and that he has re-written much of the first part of the book, in the light of the new data thus obtained.

The first chapter is an elementary explanation of the principles of the steam injector. The treatment of the subject is lucid and elementary, so far as it goes, but it omits an important part of the demonstration. It shows that a small mass of steam, moving at a very great velocity, due to the boiler pressure, can impart to a several times greater mass of water a velocity greater than that with which water would issue from a jet, under the given boiler pressure, but goes no further. The rest of the demonstration is that the steam, on meeting the feed water, is condensed to so small a volume that it makes room for more steam, and does not check the velocity of the steam jet; and the volume of the condensed steam, plus that of the feed-water, is so small that it requires only a portion of the energy developed in the steam jet to inject it under the given boiler pressure.

It is a pity that, in so good a book, especially in the third edition, more care was not devoted to the proof-reading; typographical errors are numerous and glaring.

STEAM TURBINES. THEIR DEVELOPMENT, STYLES OF BUILD, CONSTRUCTION AND USES. By Wilhelm Gentsch. Translated from the German by Arthur R. Liddell. Longmans, Green & Co., London, New York and Bombay. Cloth; 375 pages, 7x10¼. Illustrated. Plates. Price, $6.

This book is directed at the layman rather than the specialist, and intended to give the ordinary engineer and even the general reader an idea of the nature, development, construction and operation of the turbine rather than to exemplify the problems connected with its design. This object might have been better achieved than by leading the reader through chapter after chapter of relicts and abortive types, the rejecta of the process of evolution through which the turbine is passing, but the author's excuse for this is that "a knowledge of the earlier attempts—however awkward these may sometimes appear—must exert an inspiring influence on the searcher; for even in the earliest times attempts were made to solve definite problems, to remove defects and to achieve success." The author proposes a new classification into "pressure" and "velocity" turbines, according to whether the pressure has or has not been translated into velocity before the steam enters the turbine. The book is excellently printed and contains 637 engravings and 19 plates.

THE THEORY OF THE STEAM TURBINE. By Alexander Jude. Chas. Griffin & Company, Limited. J. P. Lippincott Company, Philadelphia. Cloth; 288 pages; 6x9 inches; illustrated. Price, $5.

This work starts off logically with an excellent consideration of the fundamental theorems of jets impinging on stationary and moving surfaces of different shapes and at different angles of reaction and discharge. The historical chapter is mercifully short. The third chapter is devoted to the consideration of the Velocity of Steam. The usual calculations for the academic case are given; resistance to flow is considered by the aid of the temperature-entropy diagram; the critical pressure at which the flow from a higher to a lower pressure becomes constant is explained, and various experiments on the flow of steam recounted. The centrifugal effect on steam moving in a curved path is considered, and experiments illustrating pressure oscillations within turbine vane passages described.

The turbine is then classified into types with an exposition of the application of the previously considered principles, such descriptions of practical machines as are introduced being incidental to this exposition rather than descriptive of constructional differences. The Efficiency of Turbines is considered thermodynamically, a special chapter being devoted to the efficiency of the compound turbine. Turbine Vanes, Disk and Vane Friction, the Governing and the Steam Consumption of Turbines are the captions of other chapters. Several subjects such as applied thermometry and the specific heat of superheated steam are included, although their connection with the subject is but indirect. Relating rather to the constructional than to the theoretical side of the subject are chapters upon the Strength of Rotating Disks and the Whirling of Shafts. Charts of the adiabatic expansion of steam, the temperature-entropy diagram for superheated steam, and properties of saturated steam at low temperatures are appended. The book comprises novel treatment of the subjects of leakage, governing, etc., and hitherto unpublished researches on the impact of and flow of steam through buckets, and the erosion of metallic surfaces by high-velocity steam. The book is one that will appeal to the specialist or to the advanced student of the subject rather than to the beginner.

STEAM TURBINES, PRACTICE AND THEORY. By Lester G. French. Hill Publishing Company, 505 Pearl street, New York. Cloth, 418 pages, 8¾x5¾ inches; illustrated. Price, $3.

The author was editor-in-chief of *Machinery* during the active development of the turbine and up to the present state of the art, and his book is a compilation of the knowledge and data acquired in that capacity. The book commences with a chapter on Steam Turbine Principles, showing in an easily understood way how the energy of steam in a jet is arrested and appropriated by turbines of the distinctive types.

The second chapter illustrates some of the Early Steam Turbines, and is designed to acquaint would-be inventors with what has been already attempted and accomplished, and to warn them from alluring but misleading paths. Several chapters descriptive of the different types are followed by 89 pages devoted to Steam Turbine Performance, giving the results of various tests, comparisons with the performance of reciprocating engines and considerations of the effect of vacuum and superheating.

Chapter IX is a compilation of various Experiments on the Flow of Steam, followed, after a consideration of Steam and its Properties, by a chapter of Calculations on the Flow of Steam. These chapters, with that on Turbine Vanes, are necessarily somewhat mathematical, but the difficulties have been smoothed away as much as posible. A short chapter treats of Bodies Rotating at a High Speed, explaining the problems involved in balancing, and Efficiency and Design, the Commercial Aspect of the Turbine, Care and Management, Condensing Apparatus, and the Status of the Marine Turbine receive consideration in separate chapters.

The book as a whole is not above the comprehension of the average reader and will convey to its student an excellent grasp of the principles involved in turbine engineering and what has been done toward their application.

Gear wheels are more often cut too deep than not deep enough, and, as far as noise is concerned, it is worse to have the driver too deep than the driven gear. Another cause of noise may be that the cutting is not central, as shown by gears being noisy when running one direction and quiet in the reversed direction. Again, the center distance may not be right; if meshing too deep, the outer corners of the teeth of one gear may strike hard against the roots of the teeth of the other gear. Still another cause of noise may be found in the fact that the frame carrying the gear shafts is of such a form and size as to give off sound vibrations.—*The Mechanical World.*

Preparations for the N. A. S. E. Convention

On February 19 President Kelsey, Secretary Raven, and Treasurer Force, of the National Association of Stationary Engineers, with Messrs. Carson and Thomas, president and secretary, respectively, of the National Exhibitors' Association, met at Niagara with members of the local committee to arrange the program of the convention of the N. A. S. E., which will be held in that city in September next. The program, as roughly outlined and subject to change, will be a reception at the Cataract and International hotels, which will be the headquarters, on Monday evening; on Tuesday visits to the power stations, and in the evening an impromptu dance at the hotel; on Wednesday an all-day trip on the gorge route to Lewiston, and by boat to Toronto, returning in the evening. There will also be an automobile ride for the ladies, probably on Tuesday. On Thursday evening there will be a smoker at the hotel with a vaudeville running concurrently in another room for the ladies.

The social features will conclude with a ball at the Cataract on Friday evening. The opening session will be held at the Pure Food Company's building, the subsequent sessions at the hotel. The exhibition will be held in a vacant hall in the building of the Pure Food Company.

Forced and Induced Draft

In the forced-draft system the fan usually draws the air from the boiler-room through an inlet in either side of the casing. It is thence discharged to the ash-pit. In one form of installation an underground brick duct extends along the front of the boilers, connections therefrom being made to each ash-pit where a damper operated by a handle through the cast-iron boiler front is arranged to graduate the admission to the pit, or the air may be admitted through a hollow bridgewall. The latter method cannot be easily introduced except in a new boiler plant, whereas the former method may be employed for one already installed. The bridgewall-damper method has the advantage of spreading the air more evenly through the ash-pit, and of leaving the bottom of the pit absolutely clear at all times. A slight advantage may result from the heating of the air before it is discharged to the pit.

In an induced-draft installation one or two fans may be used according to the conditions. Where the size makes it desirable or a relay is required, the duplex type of fan is generally installed, both fans being driven by direct-connected engines, the fan casings extending beneath the floor, and suitable inlet and outlet connections with dampers being provided, so that either or both fans may be operated. The relatively short stack required with this system may be easily supported by these fans. Steel-plate construction lends itself to very special conditions so that induced draft may be installed in almost any location.

Personal

Werner Boecklin, well known from his former connections with the engineering departments of the "Fort Wayne" and the "Burlington" railroad systems, the Robins Conveying Belt Company and John A. Mead & Co., has joined the engineering staff of W. S. Barstow & Co., of New York and Portland, Ore.

R. A. Perry recently resigned as engineer at the Diamond Match Company's works at Chico, Cal., to take a Civil Service appointment at the Presidio General Hospital at San Francisco. Previous to going with the match company Mr. Perry held a Civil Service position in southern Arizona for several years.

W. H. Zimmerman has resigned as general manager of the DeKalb-Sycamore (Ill.) Electric Company, to accept a similar position with the Michigan Power Company, of Lansing, Mich. The last named company contemplates the construction of a new auxiliary steam plant, as well as further developments of the water power on the Grand river. It recently completed one water-power installation, which is in operation.

Correction

In the March number of POWER a very ridiculous mistake occurred in the advertisement of the American Steam Gauge and Valve Manufacturing Company, on page 83, for which that company was in no way responsible. The advertisement as originally prepared in the POWER office was correct and was passed by the American Steam Gage and Valve Manufacturing Company. When it was put in type the printer transposed the cuts of the indicator and pop safety valve. This was not noticed by our proofreader. A proof was submitted to the American Company, which returned the proof calling attention to the mistake—but in the meantime the form containing the advertisement had been printed. POWER wishes its readers to understand that the American Steam Gauge and Valve Manufacturing Company was in no way responsible for the error in the advertisement.

Engineering Societies

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

President, Frederick R. Hutton, New York, N. Y.; sec., Calvin W. Rice; house, 29 West 29th St., New York. Next meeting, New York, December, 1907.

ENGINEERS' CLUB OF PHILADELPHIA

Pres., Henry H. Quimby; sec., Walter L. Webb; house, 1122 Girard St.; regular meetings 1st and 3d Saturdays.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Navy Dept., Washington, D. C. Pres., B. C. Bryan, Commander, U. S. N.; sec. and treas., Theo. C. Fenton, Commander, U. S. N.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., M. F. Cole, Newnan, Ga.; sec., J. D. Farasey, Cleveland, O. Next meeting, Atlanta, Ga., fall of 1907.

WESTERN SOCIETY OF ENGINEERS

Pres., W. L. Abbott; sec., J. H. Warder, 1737 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., Julian Kennedy; sec., F. V. McMullin, Pittsburg, Pa. Meetings monthly.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., Arthur Williams, New York; sec. and treas., W. C. L. Eglin, Philadelphia. Association headquarters, 136 Liberty St., New York.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dr. Samuel Sheldon; sec., Ralph W. Pope, 95 Liberty St., New York. Meetings monthly excepting July and August.

ENGINE BUILDERS' ASSOCIATION OF THE UNITED STATES

Pres., A. L. Merriam, Oswego, N. Y.;

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, Robert J. Ingleson, Cleveland, O.; sec., John J. Chisholm, 664 East 134th St., New York, N. Y. Next meeting, Chicago, Ill., August, 1907.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., T. N. Kelsey, Lowell, Mass.; sec., F. W. Raven, 525 Manhattan Building, Chicago, Ill. Next convention, Niagara Falls, N. Y., September, 1907.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., A. J. Dietrich, Baltimore, Md.; Supr. Cor. Engr., Frederick Markoe, 931 N. Orianna St., Philadelphia, Pa. Next convention, Atlantic City, June, 1907.

OHIO SOCIETY OF MECHANICAL, ELECTRICAL AND STEAM ENGINEERS

Pres., Wm. T. Magruder; sec., F. W. Ballard, 104 Canal St., Cleveland, O.

MASTER STEAM BOILER MAKERS' ASSOCIATION

Pres., J. H. Smyth; sec., Geo. M. Clark, 1337 N. Maplewood Ave., Chicago, Ill.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., Robert A. McKee, Peoria, Ill. Next convention, Philadelphia, September, 1908.

Business Items

Ernst Koerting, the well known inventor of the Koerting injector, has bought the entire interest in the Schutte & Koerting Co.

The B. F. Sturtevant Company, Boston, Mass., reports recent orders for 15 generating sets, 5 economizer installations and 19 mechanical-draft apparatus.

Manufacturers of power-plant equipment who wish to have their catalogs on file in the Department of Design at Purdue University, Lafayette, Ind., should send them to J. D. Hoffman, associate professor of engineering design.

The American Engine Company, Bound Brook, N. J., sent us a very convenient map of Manhattan island and greater New York, giving locations of interesting points and also showing the route of the subway, with all stations plainly marked.

F. E. Keyes & Son, 20 Broad street, New York, recently installed 1700 horse-power of their "Advance" feed-water purifiers in the State of New Jersey. Owing to the large increase in sales of this purifier in the middle West, a branch office will be opened in Chicago in the near future.

The Link-Belt Company has opened a new branch office at 440 New York Block, Seattle, Wash, in charge of E. G. Brabrook. Increasing business in the far East, Washington, Oregon and surrounding territory, in conveying machinery, made personal representation on the Pacific coast desirable.

By locating a branch office at 321 Ellicott Square Building, Buffalo, New York, the Link-Belt Company, of Philadelphia, Chicago and Indianapolis, is in a better position to give personal attention to customers of its drive chain department in Buffalo and the neighboring territory of New York, Pennsylvania and Ohio.

E. H. Kellogg & Co., 243 South street, New York, have received a letter from George Calder, engineer, of Vicksburg, Miss., in which he says he has used "anti-corrosive" cylinder oil for seven years and finds it superior to any oil he has ever tried. He says the first cost is high, but it is cheaper in the long run.

The Russell Engine Company, of Massillon, Ohio, announces that it will hereafter be represented in the South by the Dashiell Engineering Company, 328 Candler Building, Atlanta, Ga. J. M. Dashiell, who is in charge of the new office, is a graduate of the Naval Academy at Annapolis, and has had a wide experience in engineering.

The Russell Engine Company, Massillon, Ohio, recently received from Japan an order for three large compound-condensing direct-connected engines to be used in developing large coal mines. This will make a total of twelve engines which the Russell Engine Company has shipped to Japan, for electric mining, lighting and railway service, within the past ten months.

The Ohio Blower Company, 42 Michigan street, Cleveland, Ohio, has recently constructed for the Utah Copper Company, Garfield, Utah, three 14-inch Swartwout double-angle type steam separators, built for a working pressure of 175 pounds; each weighing 3750 pounds. The above mentioned separators were only part of the order, which included 16 Swartwout separators and two 36-inch Swartwout exhaust heads.

The H. W. Johns-Manville Company, of 100 William street, New York City, recently opened a branch office at 214 Main street, Buffalo, N. Y., in charge of George A. Schmidt, formerly of the Manville Covering Company, assisted by B. F. Boscoe, whose headquarters will be in Rochester, and Harry V. Patton. This makes sixteen branch offices for this company, besides innumerable local representatives at various points.

John Wiley & Sons, 43 and 45 East Nineteenth street, New York, have just issued a vest-pocket List of Books published by them and by Chapman & Hall, Limited, of London, during 1906. The list includes new books and new or revised editions, a synopsis of each with table of contents and pertinent data being presented in convenient form for the guidance of the intending purchaser. The booklet comprises 24 pages, 3¼x6¼ inches.

The Trinity Engineering Company has been organized to conduct a general engineering and supply business, with headquarters at 107 Liberty street, New York. The president of the new company is R. H. Thomas; general manager, W. T. Wheeler. The company announces that it has a commodious machine shop equipped with facilities to handle all sorts of machine work, engine repairs and pipe fitting. Its telephone number is 4258 Cortlandt.

The Detroit Water Tube Boiler Company, Detroit, Mich., successor to the East End Boiler Works, of that city, is building for W. C. Stetson, of Seattle, Wash., owner and manager of the steamers "Queen City," "Eagle," "Galaxer," "Detective" and "Diamond," the third Detroit water-tube (marine) boiler, the other two having been placed in the steamers "Eagle" and "Queen City." The one now building is of 50 horse-power capacity.

The Power Specialty Company, 111 Broadway, New York, has been awarded a contract covering the equipment of eight Babcock & Wilcox boilers with Foster superheaters at the United States naval proving grounds, Indian Head, Md. The Foster superheater is also to be installed in the central power plants of the Brookyn and Charleston Navy Yards and the United States Soldiers' Home, Baltimore, in several thousand horse-power of Babcock & Wilcox, Atlas and Edge Moor boilers, respectively.

The Reagan Grate Bar Company, Philadelphia, Penn., reports recent installations of "Reagan" grates at the works of the Union Switch and Signal Company, Swissvale, Penn.; Water Works Department, City of Wilmington, Del. (second order); James Lees & Sons, Bridgeport, Conn. (ninth order), and Henry Bowers Chemical Manufacturing Company, Baltimore, Md. (33d order). Recent and continued orders from New England for large installations after exhaustive tests are flattering testimonials for "Reagan" grates.

The C. H. Wheeler Manufacturing Company has just issued a circular on the C. H. Wheeler underwriter fire pump. It describes in detail the difference between the "National Standard" and the "Trade Pump." The same circular describes the C. H. Wheeler auxiliary underwriter fire pump. This is practically a new pump designed by this company. It is approved by and built according to Underwriter specifications, which recommended them for the purpose of automatically keeping a pressure on the fire mains and at the same time provide a small supply of water until the large pump can be started. The Wheeler Company reports its pump department as being especially busy at the present time.

The Chicago office of the American Steam Gauge and Valve Manufacturing Company, which has been located for the past 20 years at 16 No. Canal street, has removed to new and commodious quarters at 7-9 So. Jefferson street, where with increased floor space the company will be able to carry a much larger stock of its staple goods and specialties than ever before. A large assortment of gages of every description, a complete line of brass pop safety valves, water relief valves and indicators will be carried in stock at all times. Owing to the increased business of the company, it became imperative to move into larger quarters and at the new location the company will be able to supply the trade in a very satisfactory manner.

The Jefferson Union Company, Lexington, Mass., which recently lost the wing of its factory by fire, in which were located the boiler-, engine- and tool-rooms, has rebuilt. On account of the delay necessary to install a new engine and boiler, the company contracted with the Lexington & Boston Street Railway Company for electric power. A large motor has been installed in a fireproof building just outside the factory and necessary connections made so that the company is now running and making unions as before, much sooner than if it had undertaken to install a steam plant. We learn that the company has so many orders for its goods that it has decided to run night and day for the next three months in order to serve the demands of its customers.

The business hitherto carried on by John A. Caldwell at 45 Broadway, New York, in fuel-saving devices, including the "Ados" CO_2 recorder, has been merged into the Sarco Fuel Saving and Engineering Company. This company has just been incorporated under the laws of the State of New York, with a capital of $100,000, to take over the American interests of Sanders, Rehders & Company, Limited, of London and Manchester, and to manufacture on this side their various apparatus. Commodious offices in the new West Street Building, New York City, have been engaged, and workshops and temporary offices at 81-85 Washington street, New York. Mr. Sanders, chairman of the above named firm, is the president, and Mr. Caldwell, general manager of the corporation. Selling agencies throughout the United States will be established at an early date.

Richard Thompson & Co., 128 Liberty street, New York, have added some new things to their line of specialties, one of which is induced draft for burning cheap fuel; in installing this system they put in dumping grate bars and the Thompson damper and pressure regulator. This regulator operates the forced draft system and the damper automatically so that a uniform pressure is obtained. They are also making the Thompson planimeter which gives the M. E. pressure of a diagram without figuring. This instrument is fitted up in a morocco case, plush lined and is meeting with much favor among engineers. Circular P explains the instrument. They also report an unusually large demand for their shaking and dumping grate bars as well as indicator tube cleaners, etc. Catalog sent upon application.

The Compania Minera Borda Antigua y Anexas, S. A., whose mines are located close to the Dos Estrellas mines, near Tlalpujahua, have recently placed an order with Messrs. G. & O. Braniff & Company for an electrical installation for their mines. The motors, transformers, etc., will come from the works of the Westinghouse Electric and Manufacturing Company, Pittsburg, Penn., and the pumps from the Byron Jackson Machine Works, of Oakland, Cal. There is already on the ground, being erected, a 250-horse-power Lidgerwood Manufacturing Company steam hoist, purchased before electric power entered the camp. The Westinghouse people will attach to this hoist one of their standard 150-horse-power alternating-current hoist motors, the only necessary change in the hoist being the adding of gear and pinion to adapt the drive.

The Power Specialty Company, 111 Broadway, New York, owing to increasing business, has found it necessary to again enlarge its offices. As soon as such additional space becomes available, through the completion of

the Trinity Building Annex, the company's offices will be re-arranged and enlarged so as to include the entire Broadway front of the floor at present occupied by it. Among orders recently received for the Foster superheater for installation in the various best known types of water-tube boilers are the following: Northern Colorado Power Company, Denver Gas and Electric Company, Norwich & Westerly Railroad, Haverhill Electric Company, Electric Company of America, Allis-Chalmers Company, Westinghouse, Church, Kerr & Company, Westinghouse Electric and Manufacturing Company, Ingersoll-Rand Company, each order being for upward of 5000 horse-power.

The Ridgway Dynamo and Engine Company has just completed a large addition to its works at Ridgway, Penn. The addition is a one-story building, located immediately in front of and connected to the main machine shops. It is of steel-frame construction, approximately 200 feet long and 65 feet wide, with solid brick walls and slate roof. The building will be used as an erecting shop and for storage of finished machinery. The total floor space in the works now amounts to three acres. The entire works are driven by electric motors; all of the larger machines being provided with individual motors and smaller machines being driven in groups by motors of from 10 to 30 horse-power. Compressed air at 80 pounds pressure is piped to all parts of the plant for operating molding machines, hoists, drills, hammers, etc. All parts of the shop and office are connected by an intercommunicating telephone system.

The Geo. W. Lord Company reports that its branch in Kingston, Jamaica, while suffering severely from the earthquake, was unusually fortunate in that no lives were lost. The manager, Mr. Davidson, and his assistants escaped to the street at the first alarm and although considerable damage was done to the building, only the stock on hand in sample room was injured, all goods in warehouses being saved. A great deal of small personal property was looted by the natives. The branch was especially fortunate in having in storage the only available supply of lubricating oils on the island and consequently they have been doing a land-office business in the region of the earthquake. The Kingston branch was the chief distributing office of the West Indies for the Lord Company products until this catastrophe. Since then, the branch at San Juan, Porto Rico, has had to carry the bulk of the business.

The DuBois Iron Works, DuBois, Penn., has taken over the entire business of the Lazier Engine Manufacturing Company, Buffalo, N. Y. The DuBois Iron Works has been manufacturing gas and gasolene engines for years under the patents and designs of Arthur A. Lazier, former vice-president and general manager of the Lazier Engine Manufacturing Company who has sold out his entire interests and retires from the business management and company. Peter Eyermann, one of the foremost gas engine authorities, designers, and engineers from Germany, is chief of the engineering department and under his designs and supervision, the new and improved DuBois engine will be produced. The DuBois Iron Works will continue the manufacture of its Simplex steam pump. The officers of the company are: John E. DuBois, president; W. C. Pentz, vice-president; E. A. Badger, secretary and treasurer, with I. N. Hamilton as general manager. The sales and advertising department will be in the hands of C. E. Stuart. The head office and entire management will be located at DuBois, Penn., and at Buffalo which formerly was the Lazier Engine Manufacturing Company headquarters, a branch office will be retained together with branches in the principal cities throughout the country.

Among the industrial concerns enjoying a world-wide reputation the "Koerting concern" is one which touches nearly every industry where a boiler or stack is used. E. Koerting, whose interests in this country are identified with the De La Vergne Machine Company, of New York, and with the Schutte & Koerting Company, of Philadelphia, started the firm "Koerting" in Hanover, Koertingsdorf, in 1870, and in the course of time he added to it factories in Vienna, Italy and Russia. At the present these factories have developed to such proportions that they how employ: Koertingsdorf, 3000 workmen; Vienna (Austria), 350; Sestri (Itay), 250; Moscow (Russia), 300. If we add to this the De La Vergne Machine Company, which Mr. Koerting controls, the New York plant has 700 workmen, while the Schutte & Koerting Company employs 200 workmen. This makes a total of 4800 workmen employed, and to this can be added the great number of employees in the office and selling forces, so we may safely say that in all 6000 men are busy in this enterprise. The Koertings have their own branches in nearly all of the large cities of the United States, Germany, Austria, France, Russia, Italy, England, Greece, Sweden, Denmark, Egypt and Japan. The closely allied De La Vergne Machine Company in this country has likewise branches in our main cities, such as New York, Philadelphia, Boston, Chicago, St. Louis, Cincinnati and Atlanta. The Schutte & Koerting Company has branches in New York, Boston and Chicago.

New Equipment

The Cleveland (Ohio) Electric Illuminating Company will erect an addition.

Union Ice and Coal Company, St. Louis, Mo., will build an ice plant to cost $20,000.

The Home Ice and Refrigerator Company, Kansas City, Mo., will erect a new plant.

The Landers, Frary & Clark Company, New Britain, Conn., will erect a new boiler house.

McCann Brothers, Philadelphia, Penn., are having plans prepared for an ice-making plant.

The power plant of the Huntsville (Ala.) Railway, Light and Power Company will be doubled.

The Cosmopolitan Ice Company, New Orleans, La., will, it is stated, install a $58,000 ice plant.

It is stated the Yuma (Ariz.) Electric and Water Company will double the capacity of its plant.

The city of Fremont, Nev., has voted $50,000 to build a new municipal water and light works.

The Pacific Traction Company, Tacoma, Wash., will erect a new 40,000-horse-power electric plant.

The Granite City Gas, Light and Fuel Company, St. Louis, Mo., will erect a $20,000 addition to its plant.

The Pennsylvania Railroad Company will build an ice-making plant at Elmira, N. Y., at a cost of over $13,000.

The city of Battleford, Sask., Canada, proposes to install water works, a sewerage system and an electric-light plant.

The Portland (Ore.) Railway, Light and Power Company will build another plant, with a capacity of 25,000 horse-power.

The Green River (Utah) Light and Power Company is said to have applied for an electric-light and street-railway franchise.

The Leeds (N. D.) Electric Light and Power Company is contemplating installing a public steam-heating system in its plant.

The Illinois Traction Company will build an addition to the power plant at Danville, Ill., and will install a 300-horse-power generator.

The Missouri Lead and Zinc Company, Joplin, Mo., will install new machinery and extend its power-house, expending about $25,000.

The Bloomsfield (Mo.) Electric Light and Power Company is contemplating the installation of a water-works system and a hot-water system.

The Mac Bar Paper Company, Otsego, Mich., will spend about $100,000 in improvements. The power plant will have a capacity of 600 horse-power.

The Ft. Wayne (Ind.) Electric Works is reported to have apropriated $1,500,000 for the purpose of constructing an addition and for new equipment.

The Marion (N. C.) Light and Power Company will develop 300 horse-power on the Catawba river to furnish electricity for power and lighting purposes.

Westinghouse Electric and Manufacturing Company, E. Pittsburg, Penn., has broken ground for the erection of an additional $500,000 factory building.

One new power plant and two new substations are to be built for the United Railways and Electric Company, Baltimore, Md., at a cost of nearly $500,000.

The citizens of Newton, N. C., are considering the installation of a new electric-lighting plant of a larger capacity than the present one. W. B. Gaither is manager.

The United States Steel Company will build a new 14-inch and 22-inch structural mill at Clairton, Penn., and a 1000-kilowatt generator with gas engine will be installed.

The Columbus (Ga.) Power Company has a project for developing over 200,000 horse-power on the Chattahoochee river, which will involve an outlay of several million dollars.

The Hanover & York Street Railway Company, York, Pa., has been incorporated to build an electric railway; capital, $120,000. Grier Hersh, of York, is one of the incorporators.

It is reported that the Public Service Corporation, Trenton, N. J., will erect an additional office building and double the capacity of its power plant, involving an expenditure of $500,000.

The Lyman Manufacturing Company, of Buffalo, N. Y., expects to be in the market soon for a line of machinery and tools suitable for manufacturing brass valves and malleable unions.

H. S. Jawdon, of Savannah and Ludwig Company, of Atlanta, Ga., are the engineers for the proposed water works, sewerage system, electric light plant and ice plant to be installed at Pelham, Ga.

A company is being organized to build an electric railway from Findlay to Tiffin and Norwalk, Ohio. It will be known as the Cleveland & Indianapolis Interurban Railway. P. W. Ewing, of Findlay, Ohio, is secretary.

The Stafford Springs (Conn.) Electric Light and Gas Company is planning to build a power plant on the Willimantic river at Willington, which will be operated under the name of the Willington Electric Light and Gas Company.

The Consumers' Storage and Ice Company has been organized at Alexandria, La., to establish a cold-storage and ice plant. About $60,000 will be invested. George A. Morris, Jr., Rapids Hotel, Alexandria, is at the head of the company.

The Western Power Company has been incorporated by John Krischwing, of Spokane (Wash.), and his associates with a capital of $500,000. The company is empowered to buy patents and patent rights pertaining to current motors and inventions appertaining to the development of power by current.

The Economy Light, Fuel and Power Company, recently organized by the American District Steam Company for the purpose of waging a cut-rate war on the Lockport Gas and Electric Light Company, has declared its intention to float a $250,000 bond issue to enlarge the plant at Lockport, N. Y.

The Puyallup Valley Northern Rapid Transit Company has been incorporated with a capital of $3,000,000 to operate a system of electric interurban railways from Tacoma to Puyallup, Sumner, Auburn, Kent and Seattle. F. J. Chamberlin, F. S. Martin, of Puyallup, Wash., and others, are incorporators.

The Virginia Portland Cement Company, of Fordwick, Va., is about to add 2000 barrels per day capacity to its plant, which will be one of the largest cement plants in the South. Many improvements will be made in the buildings and machinery; there will be a new power house; and the existing mill will be driven by electricity. In the power house two turbines are to be installed, and the boiler capacity, increased by 1500 horsepower, will now aggregate 2000 horse-power. The whole of the old and new machinery will be driven by electricity. The consulting engineers of this plant are W. S. Barstow & Co., of New York and Portland, Ore. Barstow & Co. have also been retained as consulting engineers for the new plant of the Seneca Button Company, at Poughkeepsie, N. Y., which is to be operated electrically throughout, and for the Bergen Point Chemical Works, Constable Hook, N. J., which are about to be equipped with a new light and power house.

New Catalogs

Buffalo Forge Company, Buffalo, N. Y. Leaflet. Blacksmith tools. Illustrated.

Termaat & Monahan Company, Oshkosh, Wis. Catalog. Marine engines. Illustrated, 52 pages, 5½x8½ inches.

Buckeye Engine Co., Salem, Ohio. Catalog. Buckeye electric blue-printing machine. Illustrated, 15 pages, 4x9½ inches.

The Hill Machine Company, Anderson, Ind. Booklet. Hill silent electric house pump. Illustrated, 12 pages, 3½x6 inches.

Wisconsin Engine Company, Corliss, Wis. Catalog No. E-3. Corliss engines. Illustrated, 57 pages, 7x10 inches.

Comstock Engine Company, 49-61 Clymer street, Brooklyn, N. Y. Circular. Graphoil steam cylinder graphite cup. Illustrated.

Chicago Pneumatic Tool Company, Fisher building, Chicago, Ill. Catalog. Franklin air compressors. Illustrated, 116 pages, 6x9 inches.

Charles Ross & Son Company, 148-156 Classon avenue, Brooklyn, N. Y. Catalog No. 5. Paint machinery. Illustrated, 64 pages, 6x9 inches.

Crandall Packing Company, Palmyra, N. Y. Catalog and price list. Packings, gaskets, pump valves, etc. Illustrated, 80 pages, 5½x8 inches.

Wellman-Seaver-Morgan Company, Cleveland, Ohio. Booklet. "What We Do in Mining and Power Machinery." Illustrated, 24 pages, 4x9 inches.

W. H. Pearce & Co., 269 Dearborn street, Chicago, Ill. Circular. Crescent expansion joints, anchors and line traps for steam or hot water. Illustrated.

Stanley-G. I. Electric Manufacturing Company, Pittsfield, Mass. Bulletin No. 615. G. I. flush receptacle and attaching plug. Illustrated, 4 pages, 6x9 inches.

Stanley-G. I. Electric Manufacturing Company, Pittsfield, Mass. Bulletin No. 618, superseding No. 37. Types L 4 and L 14 arc lamps. Illustrated, 12 pages, 6x9 inches.

Boston Belting Company, 256-260 Devonshire street, Boston, Mass. Booklet. Forsyth combination metal insertion sheet and tubular gasket packings. Illustrated, 3½x6 inches.

Waterbury Farrel Foundry and Machine Company, Waterbury, Conn. Catalog, Section "A." "Cold Process" automatic nut, bolt, rivet machinery. Illustrated, 77 pages, 6x9 inches.

Watson & McDaniel Company, 146 North Seventh street, Philadelphia, Pa. Catalog. Steam specialties for power plants and steam heating. Illustrated, 52 pages, 3½x6½ inches.

Truxal-Painter Manufacturing Company, Chattanooga, Tenn. Catalog. Painter hoist for hoisting brick, mortar and concrete in wheelbarrows. Illustrated, 7 pages, 6x9 inches.

The Jeffrey Manufacturing Company, Columbus, Ohio. Jeffrey chain catalog No. 60. Elevating, conveying, power-transmitting machinery and chains. Illustrated, 376 pages, 6x9 inches.

Materne Manufacturing Company, St. Louis, Mo. Catalog and price list of high-pressure valves, cast-iron fittings, supplies for steam, gas and water. Illustrated, 272 pages, 5½x7½ inches.

The Direct Separator Company, Syracuse, N. Y. 1906 Catalog. Steam separators, oil separators, ammonia separators, boiler separators, exhaust heads, steam traps. Illustrated, 40 pages, 6x9 inches.

H. W. Johns-Manville Company, 100 William street, New York. Booklet. Manville fire extinguisher. Illustrated, 3½x6 inches. Booklet. Portland sectional conduits. Illustrated, 12 pages, 3½x6 inches.

Westinghouse Electric and Manufacturing Company. Circular No. 1138. Direct-current motors. Illustrated, 10 pages. 7x10 inches. Circular No. 1139. Starting and field rheostats. Illustrated, 18 pages, 7x10 pages.

Ingersoll-Rand Company, 11 Broadway, New York. Bulletin No. 2008, Imperial motor hoists and stationary motors. Illustrated, 32 pages, 6x9 inches. Catalog X 36. Imperial Type Ten air compressors. Illustrated, 40 pages, 6x9 inches.

Help Wanted

Advertisements under this head are inserted for 50 cents per line. About six words make a line.

WANTED—First-class engine erectors; permanent positions; large concern. Address Box 218, POWER.

WANTED—Experienced draftsman for stoker design. State experience and salary expected. Box 217, POWER.

SALESMAN WANTED for the sale of condensers. Schutte & Koerting Co., 12th & Thompson Streets, Philadelphia, Pa.

WANTED—Thoroughly competent steam specialty salesmen; one that can sell high-grade goods. Address "M. M. Co.," Box 215, POWER.

WANTED—High-class designing engineer for rotary valve engines. Salary $5000 to $10,000 per year to man who has had definitely successful experience. Box 191, POWER.

WANTED—A concern manufacturing mechanical rubber goods require, for their steam packing department, an energetic salesman, who has had some experience in selling steam and oil packing, gaskets, etc., to the trade. Address giving full particulars, as to previous experience, with references; also, state compensation required. Packing, 202, POWER.

Situations Wanted

Advertisements under this head are inserted for 25 cents per line. About six words make a line.

DRAFTSMAN, college graduate, with experience in the power and heating lines desires position. Box 221, POWER.

A YOUNG MECHANICAL ENGINEER would like to represent or be the agent of some manufacturer of feed water heaters of steam specialties. Box 216, POWER.

PRODUCER GAS ENGINES—Engineer experienced in design, manufacture and installation; technical graduate, desires position with manufacturer. Box 210, POWER.

CHIEF ENGINEER of large steam electric plant wishes to change; practical and technical education; expert with indicator and valve gears; excellent references; all letters answered. Box 178, POWER.

WANTED—Position as chief engineer of a medium size steam plant; practical and technical education; 9 years' experience; 27 years of age; best of references; Pacific coast or the South preferred Box 212, POWER

ENGINEER with 15 years' experience would like to change, prefer chief engineer in air compressor water works, or direct and alternating current power plant. Address "Chief Engineer City Water Works," Box 211, POWER.

YOUNG MARRIED MAN wishes position with electric railway or electric light and railway company as assistant engineer and electrician or would take charge of electric light plant in town of 5000 or over; had four years' experience in light plants. Address F. J. Weigl, Buffalo, Wyoming.

ENGINEER, requires to make change from present position; has had ten years' experience in operation of power plants, steam turbines, engines, compressors, pumps and all electrical apparatus, A. C. and D. C.; would prefer position as chief engineer of a private plant, anywhere, States or Canada; but would take a position with manufacturer or etc., where ability and sobriety counts as to promotion; am 28 years old, English born, (naturalized); married; Oxford graduate, '96; hold good references. Box 219, POWER.

Miscellaneous

Advertisements under this head are inserted for 50 cents per line. About six words make a line.

PARTY HAVING NEW stuffing box and packing desires to meet party interested in same. Box 213, POWER.

GREENFIELD VERTICAL ENGINES lead the market. Send for circular. Address Greenfield Engine Works, Harrison, N. J.

WANTED—Party with latest improved gasolene engine; have a finely equipped plant for manufacturing same. Address Box 114, Arlington, Vt.

ASBESTOS AND SHEET iron laggings for cylinders and steam chests; brass or nickel bands. John E. Widen, 87 So. Elliott Place, Brooklyn, New York.

IF YOU DESIRE to learn the latest improvement in steam boilers correspond with the Detroit Water Tube Boiler Co. See their advertisement on page 161.

ENGINES AND BOILERS, ¼ to 2-h.p. engine castings in sets. Models and general machine work. Sipp Electric and Machine Co., Paterson, N. J. Catalog 4c.

PATENTS—H. W. T. Jenner, patent attorney and mechanical expert, 608 F St., Washington, D. C. I make free examination and report if patent can be had, and exact cost.

PATENTS SECURED PROMPTLY—Highest references from prominent manufacturers. Write for Inventors' Hand Book. Shepherd & Parker, 512 Dietz Building, Washington, D. C.

WANTED—Agents to carry a staple article in steam fittings; now supplied by but one company in the United States; liberal discounts to salesmen and agents. Address Box 214, POWER.

I HAVE THREE appliances, sure money makers to those who can work them; I have no cash to put them through; would sell reasonable; for further particulars, address Box 220, POWER.

C. L. PARKER, solicitor of patents, 604 F St., Washington, D. C. Patents secured promptly and with special regard to the legal protection of the invention. Handbook for inventors sent upon request.

ENGINEERS AND FIREMEN—Send 10 cents in stamps for a 40-page pamphlet containing a list of questions asked by an examining board of engineers. Stromberg Publishing Co., 2703 Cass avenue, St. Louis, Mo.

ISAAC B. OWENS, patent attorney, 100 Broadway, New York City, specialist in oil, gas and steam engines and accessories; having made a special study of inventions and patents in this art, I am in position to afford exceptionally valuable service to inventors and manufacturers working along these lines.

For Rent

LARGE FLOORS for rent, 60x210 ft. in thriving country town near Boston and Providence. Steam and electric power, light and heat. Railroad switch. Box 222, POWER.

For Sale

Advertisements under this head are inserted for 50 cents per line. About six words make a line.

FOR SALE—Cyclopedia of electricity; five volumes, three-quarters red morocco, leather binding; books are in fine condition; used but short time; cost $19; will sell for $9. Address "R. S. M.," POWER.

FOR SALE—A slightly used engineering library. Steam, electrical, mechanical. Twelve volumes bound in three quarters red morocco leather. Cost new, $60. Will sell for $18. Address "R. L. N.," POWER.

FOR SALE—Cyclopedia of engineering; steam, electrical, mechanical; bound in three quarters morocco; four volumes; cost new, $19; are in fine condition; will sell for $9. Address "M. B. T.," POWER.

FOR SALE—Boilers, battery of two 150 h.p. Babcock and Wilcox water tube boilers for 100 lb. pressure; first-class condition; complete, all trimmings. Nathan Manufacturing Company, 416 East 106th Street, New York City.

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Potomac Electric Power Company's New Plant

Description of a Large Steam-turbine Station of Interesting Design Erected at Washington, D. C. — Concrete-block Construction

BY C. H. CLAUDY

During the past year the Potomac Electric Power Company, of Washington, D. C., has erected and is now putting into operation a new power plant, located on the Bennings road where it crosses the eastern branch of the Potomac river. The new plant, according to L. E. Sinclair, superintendent of the company, is not intended to displace existing plants, although one in Georgetown and one at the foot of Four-and-a-half street will probably be discontinued. It is more in the nature of a supplement to existing plants and it will be confined entirely to the generation of alternating current. This will be transmitted through comparatively small mains and be transformed at any or all of the nine sub-stations.

The layout of the building is similar to that of many modern turbine-generator stations. The boiler room, Fig. 4, is located at right angles to the generating room, while the switching apparatus is placed in an annex which extends the en-

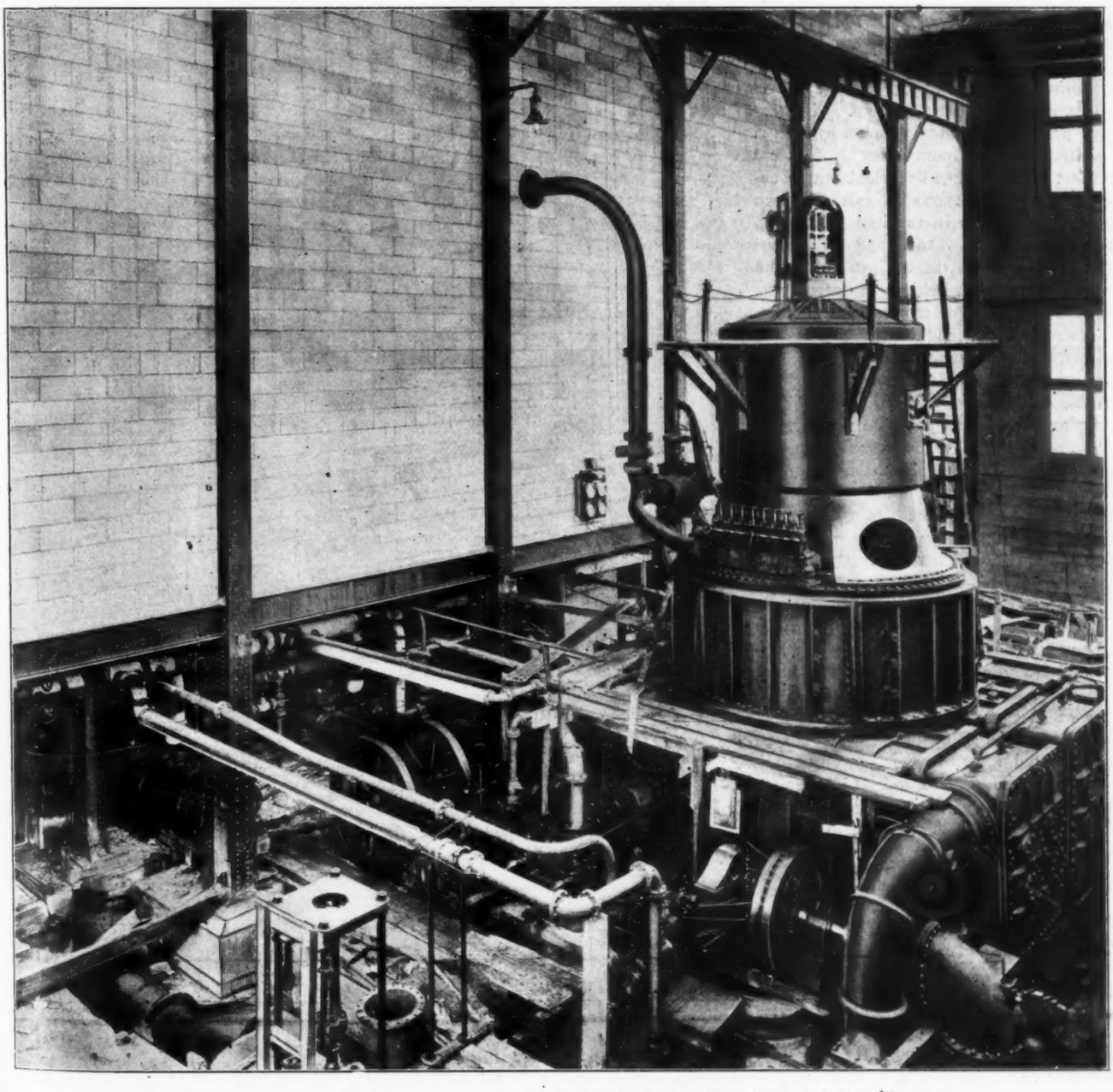


FIG. 1. 5000-KILOWATT TURBINE IN OPERATION AT POTOMAC ELECTRIC POWER COMPANY'S PLANT

tire length of the generating room. The boiler room is 164x180 feet and is arranged to accommodate four rows of boilers, forming two firing aisles. Sixteen boilers have been installed and space has been left for eight more. There are three stacks, one for every eight boilers, and two suspended steel coal bunkers with a capacity of 650 tons each. The boiler-house basement, Fig. 3, is 14 feet high, the floor being flush with the main generating room floor. Here are installed two exciter units, the boiler feed pumps, oil pumps and two house pumps, as well as oil-filtering tanks and testing tanks. The space occupied by this machinery is separated from the rest of the basement by a hollow concrete block wall, thus forming an extension of the main generating room floor. This is possible, owing to the fact that the division wall between the generating room and the boiler room does not rise from the main operating room floor, but is begun some 14 feet above and is carried on I-beams. The basement also contains the coal- and ash-handling plant, a repair shop, a storeroom, a locker and toilet room. The general plan of the plant is shown by Fig. 2.

The generating room is 164x45 feet and 70 feet high and has been laid out to accommodate three 5000-kilowatt and two 2000-kilowatt Curtis turbines. At first, however, only two 5000 and two 2000-kilowatt turbines will be installed. The annex for the switching compartments, offices, etc., is three stories, 15x164 feet.

Owing to the condition of the soil, it was found necessary to drive piles for the greater part of the footings for the engine-room walls, columns, etc. These piles are of the Raymond concrete type and are from 30 to 40 feet in length. Owing to their depth, the turbine foundations did not require piling. This was made necessary by the arrangement of the condensing-water intake and discharge tunnels, which run directly through the turbine foundations. This materially reduces the amount of concrete necessary for the foundations, and at the same time provides a most satisfactory arrangement for obtaining the circulating water. The intake and discharge tunnels have an area of 40 square feet each.

The footing of the turbine foundations is calculated for a bearing power of 1.8 tons per square foot of soil. In order to prevent uneven settlement between the different turbines, old rails have been imbedded in the concrete of the tunnel throughout the length of the building. The entire building covers an area of 29,520 square feet, and as the normal output of the plant is 19,000 kilowatts, only 1.52 square feet are taken up per kilowatt.

Up to the basement floor all wall and column footings are made of concrete, while the superstructure consists of a self-supporting steel skeleton frame. The walls are constructed of 3x1x1-foot hollow concrete blocks made by the Lake Stone Company, of Washington, D. C., and with the exception of a few of the lighter walls are self-supporting. There are some especially designed blocks, such as were required by the door and window lintels and sills. Some of the lintels have a span of 12 feet and are reinforced by ¾-inch rods. In fact, all the lintels are reinforced. The pilasters and cornices of the building are well designed and the architectural features of the structure are pleasing, as will be seen by reference to Fig. 5. The building stands close to the river bank overlooking the city and is very picturesquely located.

Equipment

As stated, there are at present sixteen boilers arranged in four rows, two boilers forming one battery. They are of the Babcock & Wilcox make, having a heating surface of 6040 square feet each, designed for 175 pounds working pressure and equipped with superheaters of 1180 square feet heating surface, capable of superheating the steam 150 degrees Fahrenheit. The boilers are provided with Roney stokers, each having a grate area of 111.8 square feet. The boiler fronts are faced with white enameled brick. The flue connections are made on the top of the boilers, four boilers joining in one breeching, thus giving two breechings for each chimney. Each boiler is provided with a hand damper, while the four main flues are equipped with dampers operated by two automatic regulators. The chimneys are of the Weber steel-concrete type and are 200 feet in hight above the basement floor, 183 feet above the grates and 163 feet above the flue openings. The internal diameter of each chimney is 12 feet. One chimney is being kept in reserve for the eight boilers yet to be installed. Between the boilers and chimneys there are two Warren-Webster open feed-water heaters, each capable of heating 200,000 pounds of boiler feed water per hour from 80 degrees up to 205 degrees Fahrenheit. In order to make up any loss caused by leakage, etc., two make-up tanks have been placed in the boiler room, each having a storage capacity of 25,000 pounds of water, and two Epping-Carpenter house pumps have been installed in that connection.

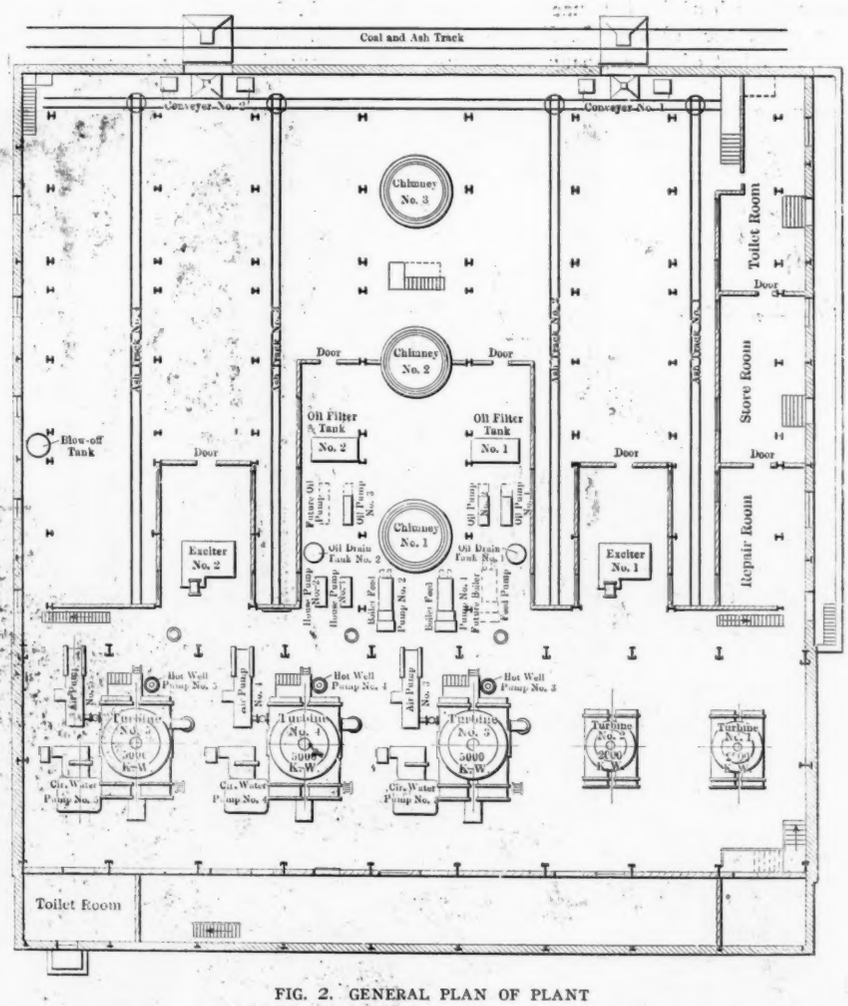


FIG. 2. GENERAL PLAN OF PLANT

The boiler feed water is supplied by two Worthington 16x10¼x16 horizontal duplex pumps. An additional pump will be installed with the full equipment of the plant. These pumps deliver to a 6-inch ring header system, from which branches lead to the boilers. The arrangement is such that either pump may supply water

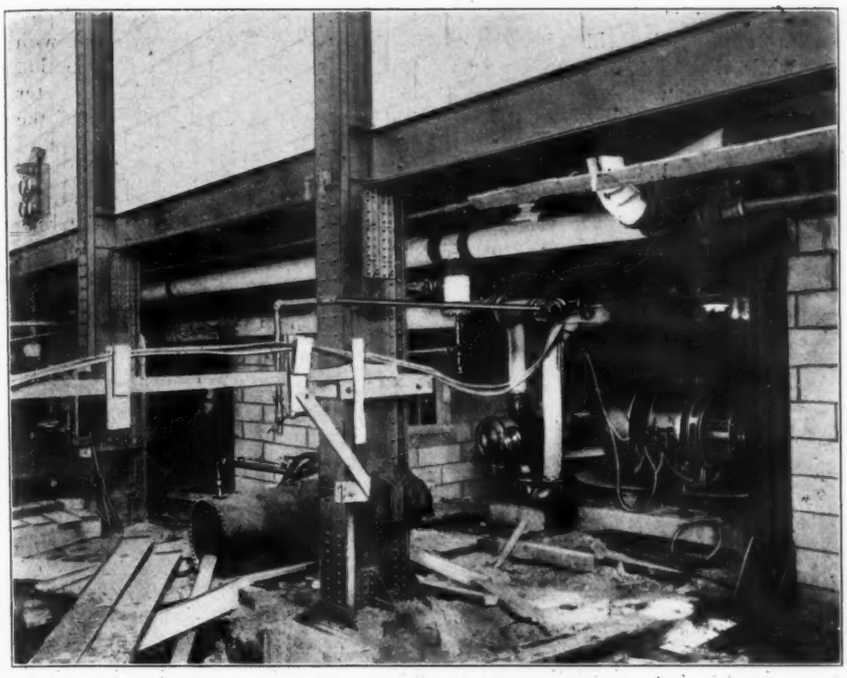


FIG. 3. THE BOILER-HOUSE BASEMENT

to any of the boilers. Branch connections to the boilers are so centrally located that the water tender may operate the valves of eight boilers from practically one point.

All steam leads from the boilers are 7 inches in diameter and are connected to the main header system by means of long-radius bends, as shown in Fig. 6. The largest size of this header is 15 inches outside diameter. The pipes are all made of steel and the fittings of semi-steel. All pipe joints are made with Van Stone flanges. The branches which lead from the header to the 2000-kilowatt turbines are of 10-inch pipe and those to the 5000-kilowatt turbines are of 12-inch pipe. The 12-inch size was chosen by the designing engineers because the use of superheated steam permits a greater velocity than saturated steam. As the turbines operate on 17 pounds of steam per kilowatt-hour, and as the volume of steam superheated 150 degrees Fahrenheit is practically 30 per cent. greater than that of saturated steam at the same pressure (175 pounds), the velocity in the 12-inch steam pipes will be approximately 7800 feet per minute.

No separators have been installed in the pipe line, but provision is made to drain the entire system by means of four 1¼-inch Squires traps, located in the basement of the boiler room. The return water is turned into the feed-water heater. A separate 6-inch steam header has been installed in the basement for the auxiliary machinery. It runs the entire length of the plant and draws steam at three points from the main header by means of 6-inch connections. This auxiliary header is provided with a drip system drained by two ¾-inch Squires traps. All steam piping, both main and auxiliary, is covered with 85 per cent. magnesia.

The 5000-kilowatt turbines are five-stage machines, designed to carry a load of 2000 kilowatts with a vacuum in the

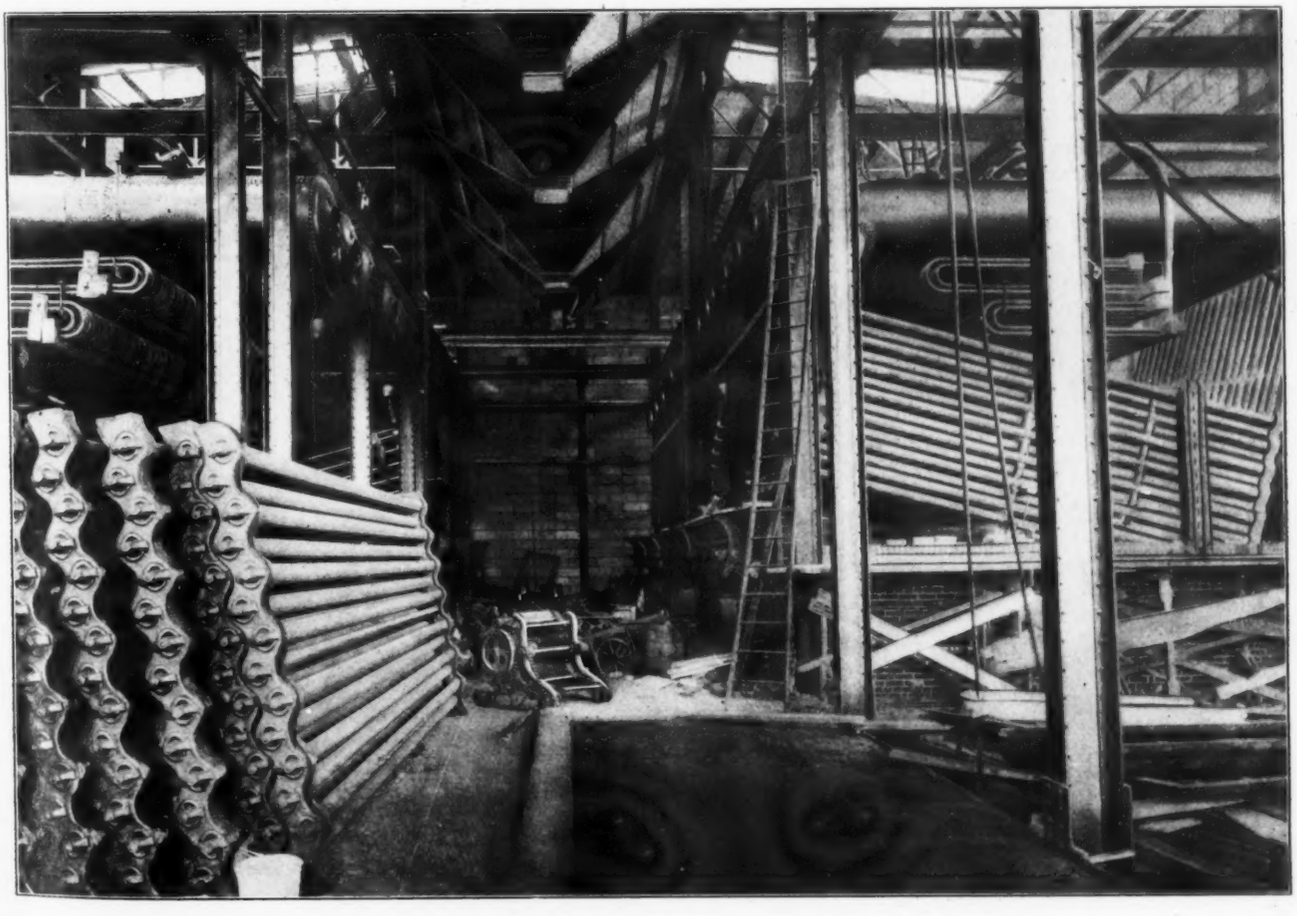


FIG. 4. BOILER ROOM OF POTOMAC ELECTRIC POWER COMPANY'S STATION

last four stages. The generators are calculated to withstand an overload of 25 per cent. for twenty-four hours, an overload of 50 per cent. for two hours and a momentary overload of 100 per cent. The 2000-kilowatt turbines are four-stage machines.

Power is at present transmitted to the city through five cables, all underground, crossing the bridge across the eastern branch of the river in iron-armored pipe. More cables will be used when the other generators are in commission. As the generators are all three-phase, twenty-five cycle, 6600-volt machines the current goes direct to sub-stations, where it is transformed to either 220 volts for lighting (three-wire), or 600 volts for railway power service.

Turbine Foundation

The foundation for the turbines is continuous throughout the turbine room and extends downward 23 feet, resting on a firm bed of gravel. Water for the condensers is brought directly into the power house through the underground tunnels leading directly to the eastern branch and below the level of the bottom of that stream, so that while the ordinary tide keeps them full, the lowest tide or drought would not affect the supply unless, indeed, the stream dried up altogether, which is hardly a possibility. Below the tunnels is a water pipe into which drains the surface water from the locality, the immense quantity of sand and gravel in the neighborhood acting as a filter, so that the water drawn from this supply pipe is pure and clear at all times. The plant needs only a 10 per cent. water supply, as all the turbines and other machinery are run in connection with condensers. The general arrangement of the intake screen chamber is shown by Fig. 7.

The cooling surface of the condensers of the 5000-kilowatt turbines comprises 20,000 square feet. The circulating water is drawn by means of a 24-inch steam-driven Worthington centrifugal pump. The vacuum pumps are of the Laidlaw-Dunn-Gordon rotative single-stage type, also steam-driven. Worthington hot-well pumps are located in the hot-well pits 5 feet 9 inches below the main operating room floor. They are vertical two-stage centrifugal pumps, and are the only pumps motor-driven. The auxiliary machinery is so grouped around one main turbine generator as to form the most compact unit system. The 5000-kilowatt unit to be installed this spring will be arranged in the same manner. The auxiliary steam piping layout is shown in Fig. 9.

The first turbine installed, a 5000-kilowatt unit, shown in Fig. 1, has been in operation since the early part of December. The two 2000-kilowatt turbines which are to be installed are not new, and are at present in operation in the company's B street plant. One of these is the unit that was exhibited by the General Electric Company at the World's Fair, St. Louis. It is provided with a base condenser, while the other unit has a separate condenser, but will, however, be redesigned in order to have the entire equipment uniform.

The auxiliary machinery of these turbines, such as the circulating-water pumps, hot-well pumps, etc., will be taken out of the existing plant. Only one of the dry vacuum pumps will be replaced by a new one.

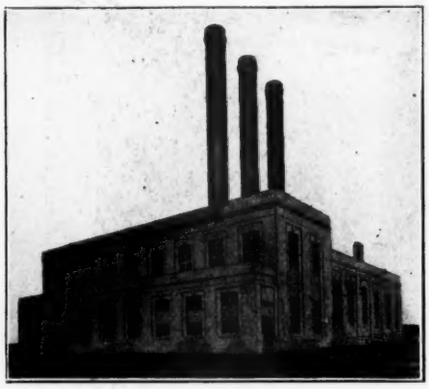


FIG. 5. EXTERIOR VIEW OF STATION

The atmospheric exhaust pipe leads in a tunnel under the generating-room floor into the boiler house, where the riser extends through the roof. These exhaust pipes, which are 24 inches in diameter on the 5000-kilowatt units, are provided with Blake relief valves. The hot-well pumps discharge into a common header which leads to the two heaters. The discharge from each hot-well pump is provided with a gate valve and a check valve, so that should one pump fail to work, the water of the other pump may not enter the pump not in operation. The suction of the vacuum pump is provided with an angle valve, so that in case of emergency the pump may be cut off from the condenser, the turbine exhausting then to the atmosphere. The discharge pipes of the dry-air pumps are connected to the exhaust risers above the relief valve. The exhaust pipes of all auxiliary machinery discharge into one common header which leads to the open feed-water heaters. The main valves are of Chapman make.

There are at present installed three Worthington pumps to supply oil to the step bearings of the various turbines and arrangement has been made for one future pump of this kind. The return oil is brought back to a Turner collecting tank, whence it is drawn by Epping-Carpenter auxiliary oil pumps and pumped into oil-filtering storage tanks, Fig. 8, each having a capacity of 1600 gallons. One filtering tank and one auxiliary pump are at present installed.

There are two 100-kilowatt, 125-volt, engine-driven General Electric exciters which are of sufficient capacity to supply the necessary current for the entire plant. These exciter units are located, as previously stated, in the basement of the boiler room, as shown in Fig. 3. The engines are of the Harrisburg type.

The first floor of the switching room in

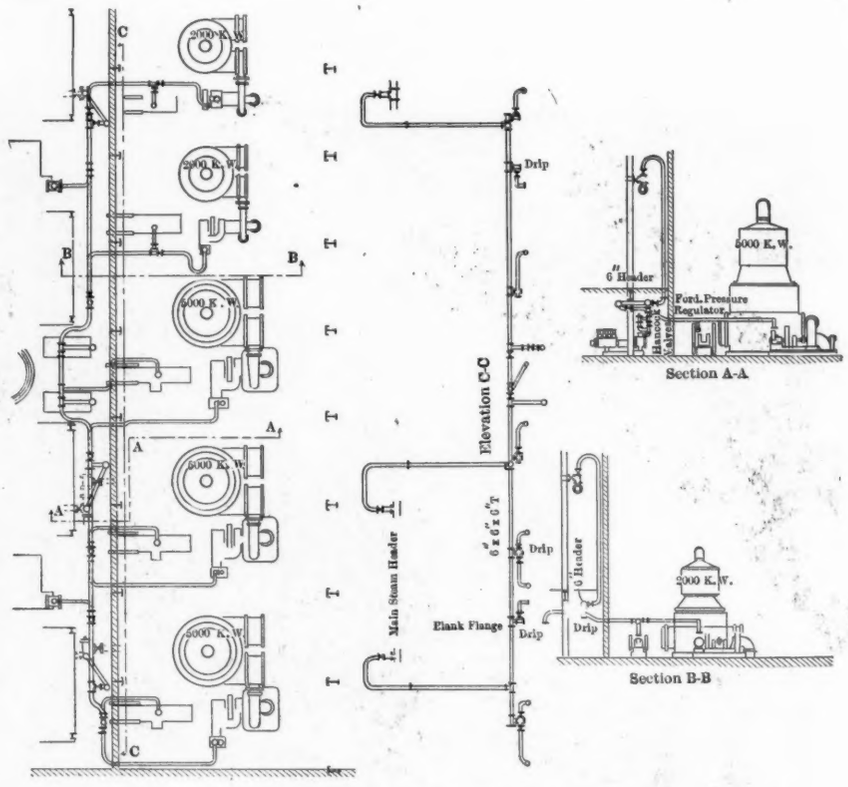


FIG. 6. AUXILIARY STEAM-PIPING LAYOUT

the annex is flush with the generating-room floor, and contains the manholes for the generator leads and the outgoing cables, which are arranged in double tile ducts on the side of the wall, whence they lead underground to the various sub-stations. Here are also installed the generator field rheostats, while at one end is located a toilet room. About 9 feet 5 inches above this floor is a gallery containing the oil switches which control the generators and the outgoing feeders, together with two bus-sectionalizing switches. These switches are of the cellular type, each pole being mounted in a separate concrete compartment, and arranged in one row, as shown in Fig. 10. Directly back of these cells are located the bus-bar compartments. The static dischargers and disconnecting switches are located between the bus-bar compartments and the wall. The switches are installed in brick compartments. The second gallery contains the switchboards. These are made of black enameled slate of dull finish, supported by a pipe framework and containing all the controlling switches, indicating lamps, instruments, etc. The entire switching compartments are separated from the main generating room by a division wall, which is provided at the upper switchboard gallery with large openings in order to overlook the generating room.

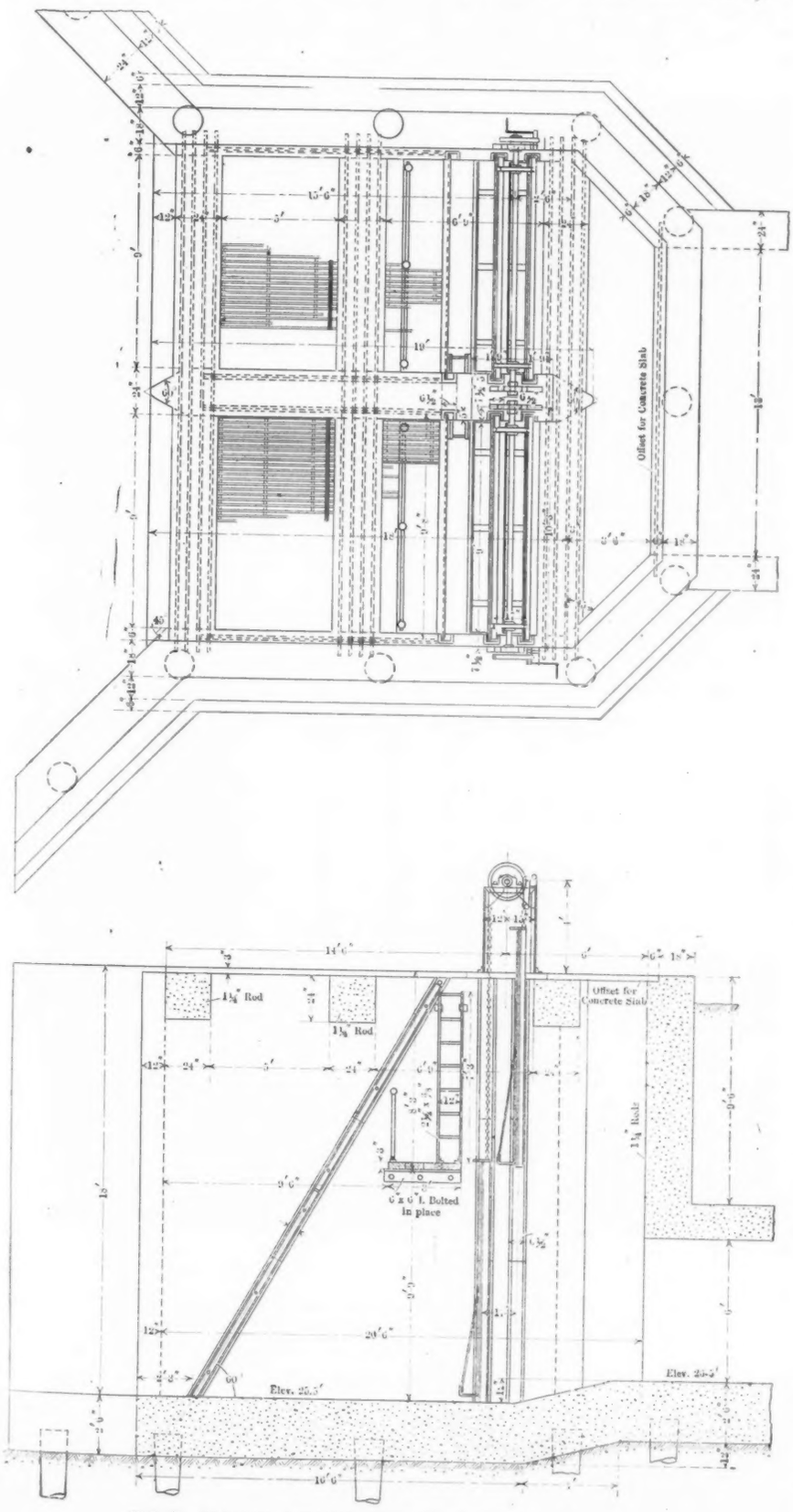


FIG. 7. GENERAL ARRANGEMENT OF SCREEN CHAMBER

Coal- and Ash-handling System

The coal is brought in by rail and unloaded by a locomotive crane in the storage yard, or may be directly dumped into receiving hoppers located at the end of the firing aisles on the east wall of the boiler room. Beneath these hoppers at the basement floor are located coal crushers from which the coal is delivered into bucket conveyers, elevating the same to the top of suspended coal bunkers, where the coal is received by two belt conveyers with a capacity of 40 tons per hour each and distributed over the bunkers.

The coal crushers are of the single roll type, and are supported on structural steel. They are driven by 25-horse-power, 25-cycle, 220-volt, three-phase alternating-current motors. The teeth of these crushers are removable. The bucket conveyers are of the McCaslin overlapping system, and are driven by 15-horse-power, 25-cycle, 220-volt induction motors, of Westinghouse Machine Company make. In order to drive the belt conveyers by the same motors, the motors are located beneath the roof of the building. Between each motor driving a bucket conveyer and the belt conveyer are inserted friction clutches in order that the bucket conveyer may be operated without running the belt conveyer. The belt conveyers are 16 inches wide and are provided with trippers.

The coal bunkers are made up of ¼-inch rolled-steel plate, and are suspended from the boiler-house columns. The capacity is 6 tons per running foot, and as there are two such bunkers, one above each firing asile, the total capacity is approximately 1300 tons. From these bunkers the coal is guided by means of chutes into the receiving hoppers of the stokers.

For removal of the ashes an industrial railway system has been installed in the basement of the plant and ashes are dumped from the ash hoppers suspended beneath the boilers into small cars and delivered into receiving hoppers near the bucket conveyer. From here they are elevated by means of the above-described bucket conveyer into suspended ash hoppers, which are located between the end

of the coal bunkers and the building wall. These ash hoppers, of which there are two, are made of reinforced concrete, and have a capacity of approximately 50 cubic yards each. From here the ashes are removed by railroad cars.

In addition to supplying power to much of the mileage of the street railways of Washington, the company lights most of the streets and does a general lighting and power-supplying business.

Wooden Fly-wheels

The *London Engineer* says that after an accident to a fly-wheel in a large European electric station the superintendent designed and had constructed a fly-wheel of wood which has a diameter of over 35 feet, and a rim width of 10 feet. The thickness of the rim is about 12 inches, and it is made up of 44 thicknesses of beech planks, with staggered joints. The boards were glued together and then bolted. The inside consists of a double wheel, the 24 spokes of which are fastened to two hubs. Spokes and hubs are of cast-iron. The wheel is operated at 76 revolutions per minute, which corresponds to a peripheral speed at the rim of 139 feet per second.

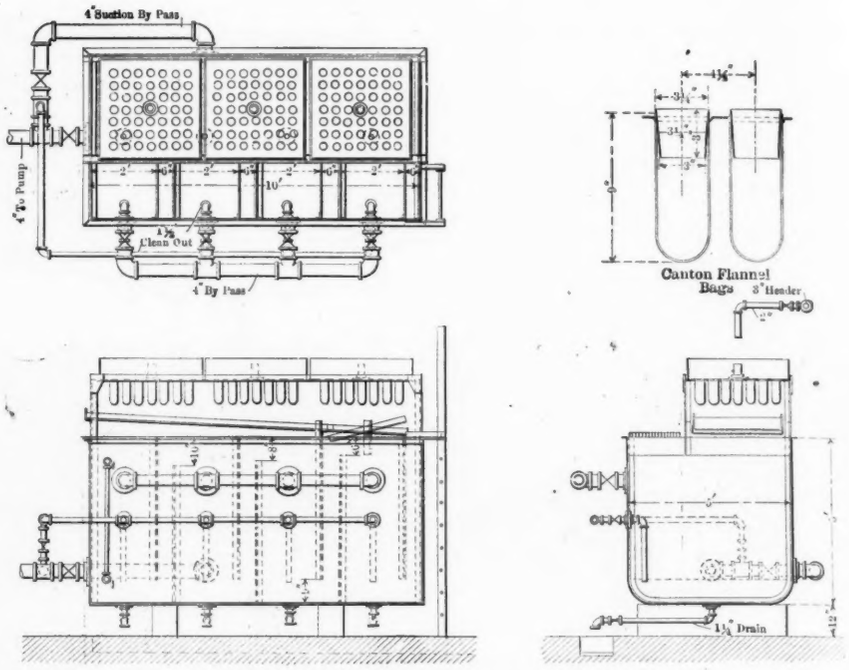


FIG. 8. CONSTRUCTION AND LAYOUT OF OIL-FILTER TANK

In the issue of April, 1892, POWER published a description of a wooden fly-wheel designed by Charles H. Manning, superintendent of the Amoskeag mills, Manchester, N. H., to replace an iron wheel which burst several months before. The wood wheel was 30 feet in diameter, 108¼ inches face and 12 inches thick. It was made up of 44 rings of western ash, each ring being in 12 sections. The weight was

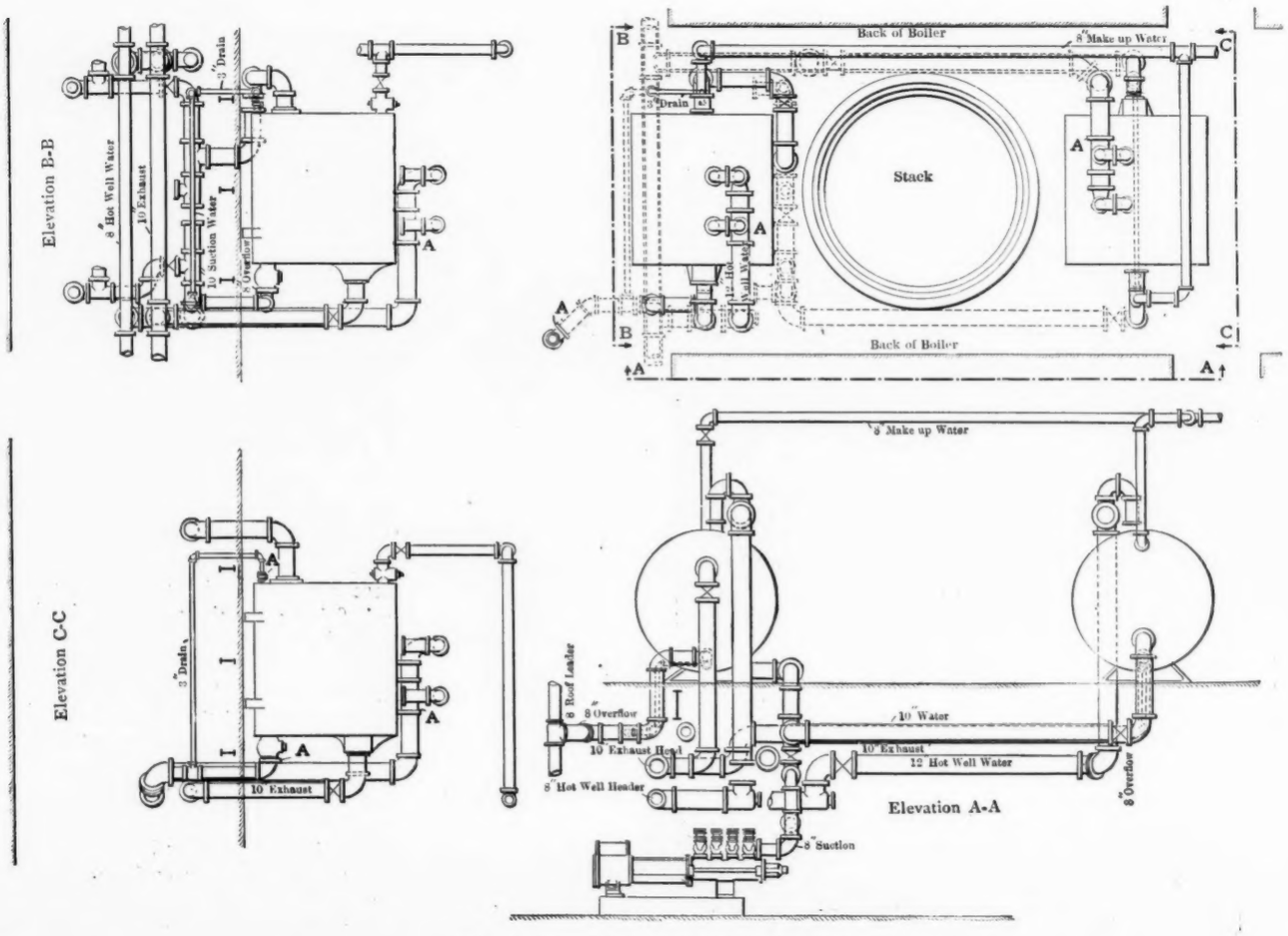


FIG. 9. PIPING FOR FEED-WATER HEATERS

a little more than 104,000 pounds, as compared with 116,000 pounds for the old iron wheel.

In March, 1893, we described a wooden fly-wheel which was built for the Willimantic (Conn.) Linen Company, to replace an iron wheel which burst in June, 1892. The new wheel was 28 feet in diameter, 110 inches face, and weighed 160,000 pounds, against the 187,000 pounds' weight of the old wheel.

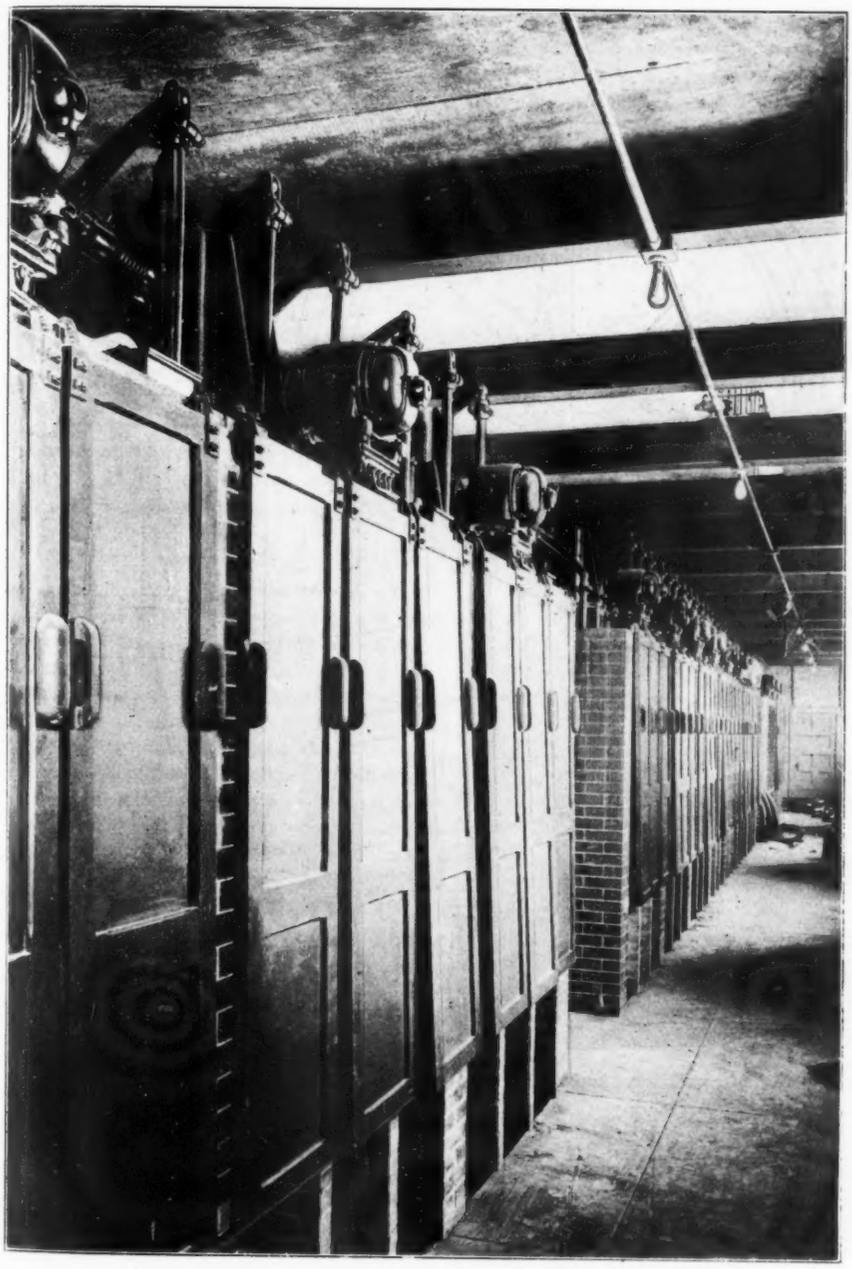


FIG. 10. A LINE OF MOTOR-OPERATED HIGH-TENSION SWITCHES

In November, 1894, mention was made by the South Bend Pulley Company of a wooden fly-wheel, 20 feet in diameter, 38 inches face, which had been running for a long time, making 145 revolutions per minute and transmitting more than 600 horse-power.

In July, 1892, it was announced that the Lawrence Corporation, of Lowell, Mass., would build a 30-foot wooden fly-wheel.

Connecticut River Dam near Brattleboro, Vt.

The Connecticut River Power Company has received charters from the Vermont and New Hampshire legislatures, and will dam the Connecticut river from Cooper's point, Hinsdale, N. H., to Vernon, Vt., five miles below Brattleboro. The river makes an oxbow bend at this point, has a rock bottom, and there is an island of rock in mid-river, across the head of which the dam will be built. The dam will be of reinforced concrete, with a fall of 26 feet and an 800-foot spillway. The drainage area of the reservoir which will be formed is 6300 square miles. A concrete canal will run from the dam to the power house on the Hinsdale side, about 500 feet from the river.

A Shutdown in a Traction Station Due to Meddling

By E. Russell

In a certain London railway power house equipped with turbines, the entire plant being wholly up-to-date, an accident happened a few months after the inauguration of the service, resulting in a complete shutdown of 12 minutes' duration, it being the only interruption of consequence in the history of the plant.

It was about 6:15 p.m., with the evening peak nearly at maximum, when the boilers began to prime badly and considerable water containing a sediment with which this station is troubled went over into the turbines. The machines themselves were uninjured by this, but the deposit caused a relay plunger to stick in the steaming position, so that the turbine speeded and brought the safety governor into action, shutting the machine down.

Before this governor could be restored to its original position it was necessary for the speed to drop about 10 per cent. below the normal running of 900 revolutions per minute. The contractor's fitter, who was looking after the machine, knew his job and at once stood by the runaway governor to adjust it when the speed was sufficiently low. At the same time he notified the railway company's engineman, who was supposed to be attending the stand-by machine, to pull a vacuum and get everything clear for running up, the intention being to keep the load on the disabled turbine by running on the stop valve, paralleling and shutting down; for, of course, with the load still on, the machine would pull up long before the other machine could be got under way, as the load at that time would be about 2600 kilowatts.

Before the fitter had a chance to do anything, however, members of the contractor's staff representing other departments, anxious to show their authority over the fitter, commenced to meddle and give orders, with the result that the safety governor became disabled temporarily. Then they proceeded to the stand-by, but in endeavoring to start the dry-air pump they had bent one of the Corliss valve spindles, so they were unable to accomplish anything. Next they tried to use the other spare machine, but as the air pump in this also gave trouble, through their not understanding the gear, they hurried back to the original machine, where the fitter had remained. By this time, having been left to himself, the fitter had managed to adjust the runaway governor and free the relay plunger, and was running up to speed once more, the machine being "on load" again in just 12 minutes from the beginning of the trouble. If the officials from the other branches had not interfered, the current would not have been cut off, probably.

A High-efficiency Fly-wheel Joint

It has come to be accepted as a matter of course that the construction of a segmental fly-wheel which shall have the same strength in the rim as a wheel cast in one piece is an impossibility. Designers have insensibly compared a fly-wheel joint with a boiler-shell joint and considered that a joint having an efficiency of 100 per cent. is impossible in both cases.

As a matter of fact, this view, as regards fly-wheels, is entirely erroneous, expedients being possible with fly-wheels which are impossible with boiler shells; and by the application of these expedients it is entirely feasible to make a fly-wheel of which the joints shall be the strongest part.

Efficiency of the Bolted Joint

The first to call attention to this fact was John Fritz. Some years ago Mr. Fritz described in a paper read before the American Society of Mechanical Engineers a fly-wheel consruction which he had used in many cases at the Bethlehem Steel Works, and in which this result was accomplished. A different application of the same principle has been invented by H. V. Haight, chief engineer of the Canadian Rand Drill Company, Sherbrooke, Quebec, and this construction will be found illustrated herewith. Mr. Fritz's construction was intended for segmental wheels and involved a hollow rim, the making of which for a wheel in halves would involve difficulties in the foundry. Mr. Haight's construction, on the other hand, is equally applicable to segmental wheels or to those cast in halves, and it is precisely this latter pattern of wheel which stands most in need of improvement. The common form of joint used with wheels in halves—a pair of lugs cast within the rim and connected by a bolt, the joints being midway between the arms—is altogether the worst construction of all. W. H. Boehm, manager of the fly-wheel insurance department of the Fidelity and Casualty Company, who has given much attention to this subject, concludes, after analyzing the experiments by Professor Benjamin, that this pattern of joint cannot have an efficiency above 25 per cent., and that in many cases the efficiency goes as low as 20 per cent. Mr. Boehm has tabulated the efficiencies of various forms of joints and concludes that with those in common use the best efficiency that can be obtained is 60 per cent.; although in some recent and very expensive joints, which are reinforced by staggered plates riveted to the sides of the rim, this figure is exceeded.

Will the Joint Be Adopted?

Mr. Haight has supplied a joint of cheap and feasible construction, and with this before them there is no reason why manufacturers should continue to use the dangerous joints which are now so common. Professor Sweet has repeatedly said that it is almost impossible to give away a good idea. Here is a construction by which wheels may be made five times as strong as some, four times as strong as many, and twice as strong as most others. Is it possible that Professor Sweet's dictum shall hold and that Mr. Haight's construction shall pass to the limbo of forgotten things?

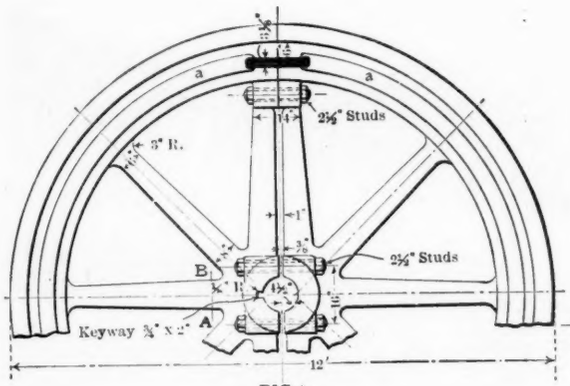


FIG. 1

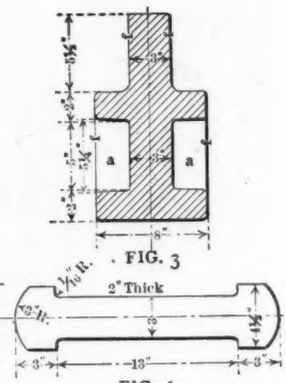


FIG. 4

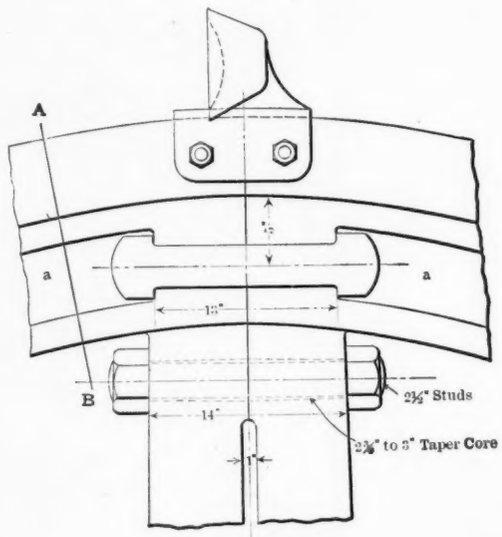


FIG. 2

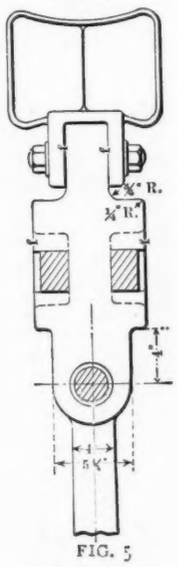


FIG. 5

A FLY-WHEEL JOINT WHICH IS AS STRONG AS THE SOLID RIM

The Cause of the Weakness of Common Joints

The reason why ordinary joints are weaker than the parts which are joined is that those parts are cut away to provide room for the joining pieces. This cutting away reduces the strength of the joined parts, and the greater the provision for strong connecting pieces, the weaker becomes the rim section, the result being that the designer aims to balance the strength of the two parts, and when he has done this he concludes that the joint is the strongest possible. Mr. Haight's plan is to cut away the rim section throughout its circumference, carrying the section which is imposed by the joining pieces all the way around the rim, the result being that the rim is not weakened by making provision for the joining pieces. Since the strength of the rim at the joined end is then the same as that at other points in its circumference, it follows that there is no limit to the extent to which provision can be made for the joining pieces; and it becomes necessary only to make those pieces as strong as the section of the rim in order to provide a joint which has the same strength as the rim. As a matter of fact, there is no difficulty in making the joining pieces stronger than the rim, or in making the section of the rim at this point stronger than that at other points of its circumference, and we hence have the unique fact that this joint may have an efficiency exceeding 100 per cent.

Jointed Wheels Stronger Than Solid Ones

Not only is it possible to design a joint of this type, which shall be stronger than the solid section of the rim, but it is altogether probable that a wheel can be designed on these lines which shall be

stronger than a wheel cast in one piece. The usual form of wheel-rim section involves a spongy center, which adds its due quota to the weight of the rim and to the strains to be carried by the rim section, while it adds very little to the strength of the section. Mr. Haight's construction involves a ribbed form of rim, by which this spongy center is largely eliminated, and hence the section of his castings should be stronger than that of the usual form; and, inasmuch as the link can then be made of the same strength as the casting, the conclusion would seem to be inevitable that the wheel as a whole should be stronger than a solid wheel. It is very greatly to be desired that this construction should be tested by Professor Benjamin in comparison with a solid wheel, and I certainly hope he will see fit to make it a subject of experiment.

Mr. Haight's joint was devised for use in connection with the runner of a Pelton water wheel. In such a case, of course, the buckets add to the strain upon the wheel while they add nothing to its strength, and, the speed being high, a high-efficiency joint became necessary. The joint shown is hence a distinct invention which was made in an emergency case—a consideration which will not diminish the reader's admiration for it.

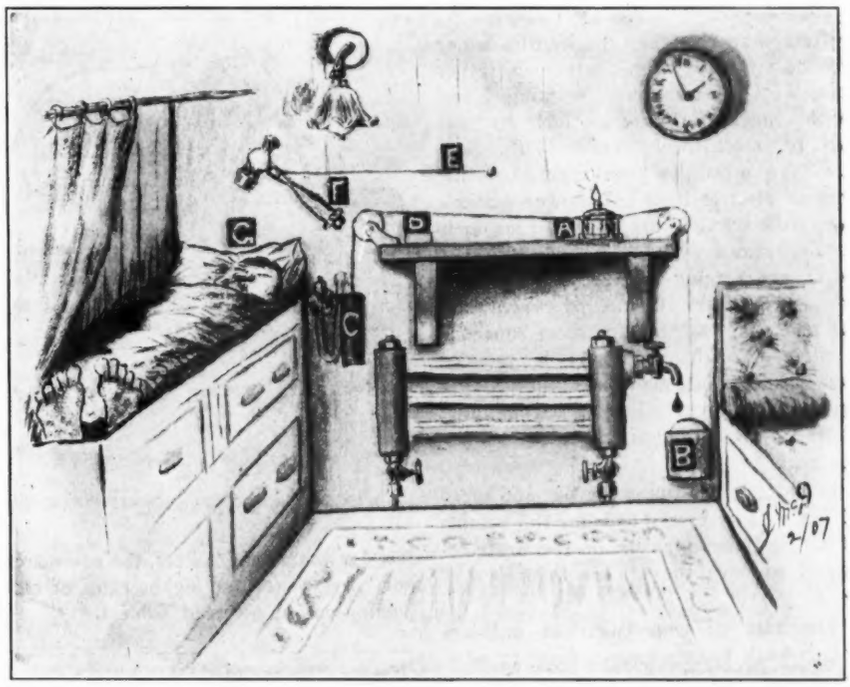
The Design in Detail

Fig. 1 shows a side elevation of one half the wheel, and the other views give the joint in detail. The simple removal of the metal which would ordinarily occupy the channels *a a* and its addition to the other parts of the section, where it acts to strengthen the section at the link as well as the remainder of the wheel, accomplishes the seemingly impossible. There could scarcely be a simpler and neater solution of a difficult problem. Fig. 3 is a normal section of the rim, and Fig. 5 a section through the joint, showing the links; the Pelton bucket being shown in Figs. 2 and 5. The extreme diameter of the wheel is 12 feet, and the normal speed is 100 revolutions per minute. With the load removed and the wheel running free, the extreme possible speed of a Pelton wheel is twice the normal, that is, in this case, 200 revolutions, giving a rim speed of 7500 feet per minute. It was desired to have the wheel safe under this condition, and, with the load due to the buckets added, calculations show the strain upon the normal rim section to be 1470 pounds per square inch, and upon the links 8800 pounds per square inch. These strains will be seen to be entirely safe, although, had it been thought necessary, the links could have been enlarged and the strain upon them reduced in the manner already described.

The manner of dividing the wheel with the joints at the arms instead of half-way between, as is customary, will be seen to be strictly correct, as the cantilever action of the unsupported ends of the segments is eliminated, this cantilever action being the chief reason for the very low efficiency of the common bolted joint. Experiments by Professor Benjamin have shown that it is this cantilever action which determines the speed at which wheels of the common type let go. The bolts through the arms and within the rim assist the action of the links, somewhat, although the reason for their use lies in the machining of the wheel of which the rim had to be turned in order to accommodate the buckets. Were the bolts omitted from the finished wheel no particular harm would be done.—F. A. H., in *American Machinist.*

Another Steam Trap

Having seen Dixie's account, at page 487, Vol. 29, Part 2, of how his chief engineer detected the stopping of the fans to save steam, I read it to the boys in the mess-room, and gave it as my opinion that it was a very ingenious contrivance.



ANOTHER STEAM TRAP

Thereupon, Ananias, the third assistant, related a similar narrative about a chief with whom he had sailed. Like the engineers of Dixie's ship, they did everything possible to save steam and get revolutions, and one of the methods adopted during the night watches was to shut off heaters, which were coils of pipe inclosed in large boxes through which the air was driven by electric fans, and which on a passenger ship *do* run away with a lot of steam. The apparatus, as seen in the sketch, consisted of a shelf at the ends of which were two pulleys, over which ran two cords, one end of each cord being attached to a small oil lamp *A* situated on the shelf near one end. The other ends of the cords suspended a weight and a tin can, respectively. Under the shelf was the heater or radiator for the chief's room, on which there was a small drain cock that could be set to drip into the tin can *B*, which it kept about three-quarters full, there being a small hole in the bottom to allow the water to run or drip away.

The moment the heaters were shut off the drip stopped, the can emptied itself, and the weight *C* overcoming the now empty can descended, dragging the lamp to the other end of the shelf against the stopper *D*, which brought the flame of the lamp against the cord at *E*. Burning it through released the hammer which (like the sword of Damocles) was suspended over the chief's head, and had a fulcrum on the electric-lamp switch *F*. It struck the chief on the head *G* (ee whiz!), and at the same time switched on his light and rang the bell to the engine room. This put the chief in a frame of mind fit to discuss the subject fluently and forcibly with the victim who should answer the bell.

The idea is a very simple one, although not easy to "fake," and as it is not patented or protected in any way any brother engineer is welcome to profit by the idea and have it installed at practically no cost or trouble; for I have it from Ananias himself that it was never known to fail, and could not be fixed with a little glue like Dixie's cigar box.—Jas. McDonald, First Assistant Engineer, S. S. "Samland," in *American Machinist.*

Fans for Draft, Forced and Induced—I

Directions for Calculating Proportions and Capacity of Fans, the Horse-power Required, and the Volume of Air Delivered

BY CHARLES L. HUBBARD

Mechanical draft may be used either as a substitute for a tall chimney or for increasing the power of a chimney already in use. With mechanical draft the boilers may be run at their maximum capacity without regard to outside weather conditions, and as the draft is independent of the temperature of the chimney gases it allows of their being cooled to a comparatively low degree, thus making available a certain quantity of heat which may be utilized in an economizer, and a saving of ten per cent. or more in fuel is often accomplished in this way. Another advantage is that poorer grades of fuel may be used than is possible with a natural draft.

Tests show that the capacity of a boiler may be increased from thirty to forty per cent. for continuous working, and much more for short runs by the use of mechanical draft. This feature is especially welcome in the case of electric light and power stations where the maximum quantity of steam is required for only a few hours per day. Of course, forcing a boiler beyond a certain limit lowers its rate of evaporation per pound of coal, but for short runs this is often more than offset by the reduced first cost of the plant and the smaller operating expenses. Ease of regulation is another advantage of considerable importance.

In order to determine the size and speed of fan, it is necessary to know the volume of air and the intensity of the draft required to burn the grade of fuel to be used.

The rate of combustion in ordinary power work usually ranges from 15 to 30 pounds of coal per hour per square foot of grate surface. In marine and locomotive boilers it is much higher, varying from 30 to 50 pounds in the former and from 60 to 125 in the latter. If the rate of combustion be increased in the case of a boiler steaming at maximum efficiency with natural draft, while the grate surface remains unchanged, the rate of evaporation per pound of coal will be reduced; but if the grate area be reduced so that with a higher rate of combustion per unit of surface the total quantity of fuel burned per hour remains unchanged, a certain increase in efficiency will be obtainable. This is due principally to the deeper fire, which may be carried with a forced draft, thus causing the air to come in contact with a greater amount of fuel and affording a better opportunity for more perfect combustion. This reduces the quantity of air required, produces a hotter fire and radiates more heat to the boiler surfaces.

Experience has shown that under ordinary conditions a combustion of about 0.3 of a pound of coal per square foot of boiler heating surface seems to give the most satisfactory results. This should be borne in mind as a guide when applying forced draft to a boiler. As the rate of combustion is increased the total amount of coal consumed should be kept approximately the same by reducing the size of the grate, as indicated above. Assuming 15 square feet of heating surface per horse-power for tubular boilers, and 10 feet for water-tube boilers, the grate areas per horse-power for higher rates of combustion will be given in Table I.

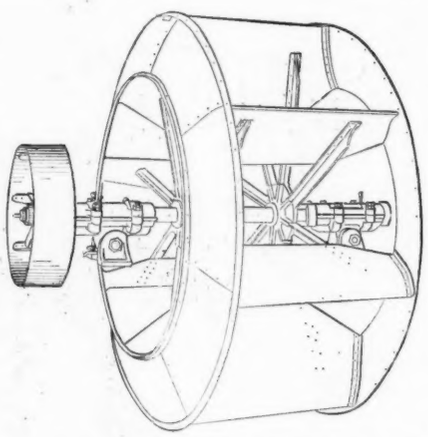


FIG. I. THE ROTOR OF A CENTRIFUGAL FAN

TABLE I.

Pounds of Coal Burned per sq.ft. of Grate per hr.	Square Feet of Grate per Horse-power.	
	Tubular Boilers.	Water-tube Boilers.
15	0.30	0.22
18	0.25	0.19
21	0.22	0.16
24	0.20	0.14
27	0.17	0.12
30	0.15	0.11

The draft or pressure required will vary in different cases, depending upon the quality and fineness of the fuel, the length and size of the smoke passages and the number of bends in the pipe.

In ordinary practice with stationary boilers, it usually runs from 1 to 2 inches of water column. Table II, tabulated by Dr. R. H. Thurston, gives the pressure required to properly burn different grades of fuel.

TABLE II.

Kind of Fuel.	Pressure Required, in Inches of Water Column.
Good steam coal.	0.4 to 0.7
Ordinary slack.	0.6 " 0.9
Very fine slack.	0.7 " 1.1
Semi-anthracite.	0.9 " 1.2
Anthracite slack.	1.3 " 1.8

If the smoke passages are long, or the flow of gases through them is restricted in any way, the pressures given in Table II should be correspondingly increased.

There are two systems of mechanical draft commonly applied to stationary boilers, known as forced draft and induced draft.

The forced draft or blower system is the one most easily applied to existing plants when it is desired to increase the draft for burning cheaper grades of fuel. The air is usually introduced through openings in the bridge or in the bottom of the ash-pit. Care should be taken to distribute the air supply as evenly as possible in order to prevent holes being blown through the fire. This may be accomplished by the use of deflecting dampers, which throw the air downward as it enters the ash-pit. The pressure within the ash-pit and furnace tends to blow ashes and flame into the fire room if forced too hard, but under a moderate draft little trouble will be experienced from this cause. Care should always be taken to close the damper in the blast pipe before opening the furnace or ash-pit doors.

With the induced-draft system, a partial vacuum is produced within the furnace and its effect is, therefore, practically the same as that of a chimney. It is more easily controlled than other systems, and as the leakage is inward all inconvenience from flame and dust blowing out at the fire doors is avoided. A specially constructed fan, with reliable means for keeping the bearings cool, is required for this system, as the temperature of the gases passing through it ranges from 300 to 500 degrees Fahrenheit, or even more.

Theoretically, about 12 pounds of air are required to burn one pound of carbon. In practice, however, with natural draft, it is necessary to supply about twice this amount to secure complete combustion,

owing to the difficulty of bringing the air into contact with the gases distilled from the coal. With forced draft and deeper fires, the quality of air supplied may be reduced to about 18 pounds per pound of coal, which is equivalent in round numbers to 230 cubic feet. This reduction

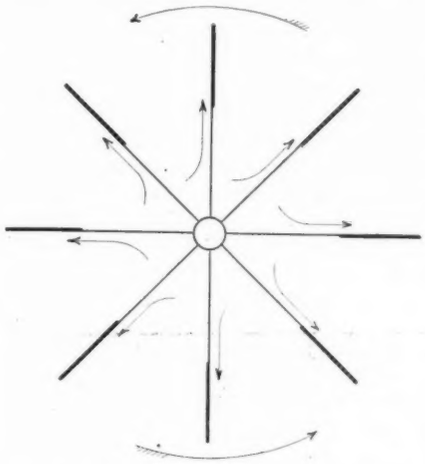


FIG. 2. THE ACTION OF A CENTRIFUGAL FAN

should, theoretically, increase the efficiency of the boiler about ten per cent.

The centrifugal fan or blower is commonly used for mechanical draft. This consists of a number of straight or slightly curved blades extending from an axis as shown in Fig. 1.

When the fan is in motion the air in contact with the blades is thrown outward by the action of centrifugal force and de-

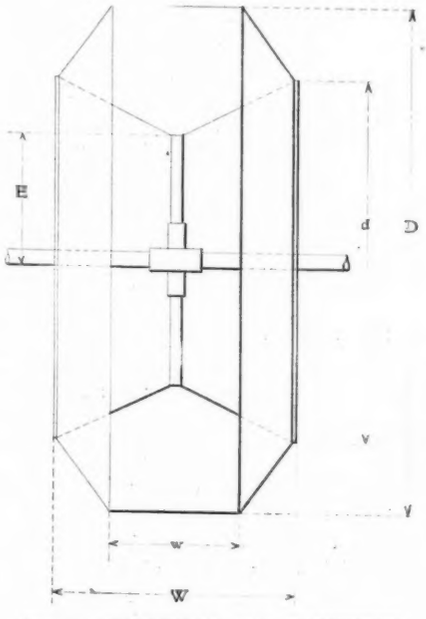


FIG. 3. CENTRIFUGAL FAN PROPORTIONS

livered at the outer circumference or periphery of the wheel. A partial vacuum is thus produced at the center of the wheel, and air from the outside flows in to take the place of that which has been discharged. Fig. 2 illustrates the action of a centrifugal fan, the arrows showing the path of the air.

This type of fan is usually inclosed in a steel-plate casing of such form as to provide for the free movement of the air as it escapes from the periphery of the wheel. An opening in the circumference of the casing serves as an outlet into the distributing ducts to the furnace or to the chimney in the case of induced draft.

The general form of fan-wheel is shown in Fig. 1, which represents a double spider wheel with curved blades. Those over four feet in diameter usually have two spiders, while fans of large size are often provided with three or more. The number of floats or blades commonly varies from six to twelve, depending upon the diameter of the fan. They are made both curved and straight; the former, it is claimed, run more quietly; but if curved too much, will not work so well against a high pressure as the latter form.

Fig. 3 represents a fan-wheel in diagram, and shows the principal dimensions to be considered.

The following proportions are averages taken from fans of different sizes as made by several manufacturers for general work:

$d = 0.68\ D$
$E = 0.25\ D$
$W = 0.52\ D$
$w = 0.8\ W$

where D = diameter of wheel
d = diameter of inlet to wheel.
E = distance from center of fan to heel of blades.
W = width of fan at inlet.
and w = width of fan at periphery.

These proportions, as already stated, do not represent those of any particular make, nor follow any fixed rule, but are general averages as taken from the catalog dimensions of several well-known manufacturers.

The size of a fan is commonly expressed in inches, and indicates approximately the hight, in inches, of a fan with full housing. The diameter of the wheel is usually expressed in feet, and can be found in any case by dividing the size in inches by 20.

For example: a 120-inch fan has a wheel $120 \div 20 = 6$ feet in diameter.

While the probable action of a fan may be quite accurately computed when operating without external resistance in free air, it becomes somewhat complicated in the case of a fan working under the conditions of actual practice.

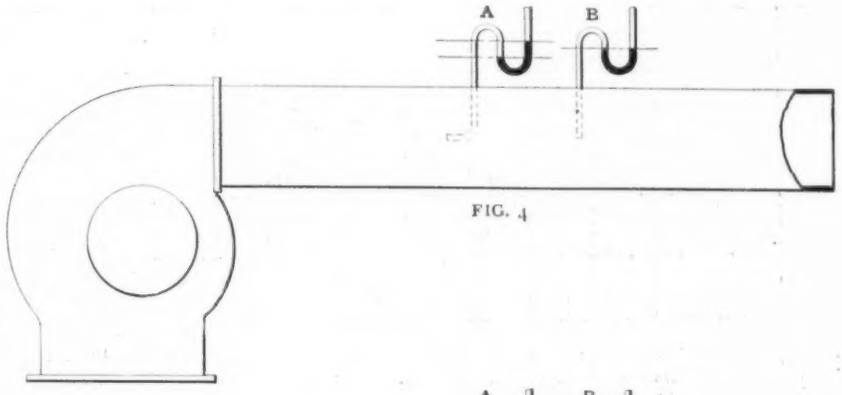


FIG. 4

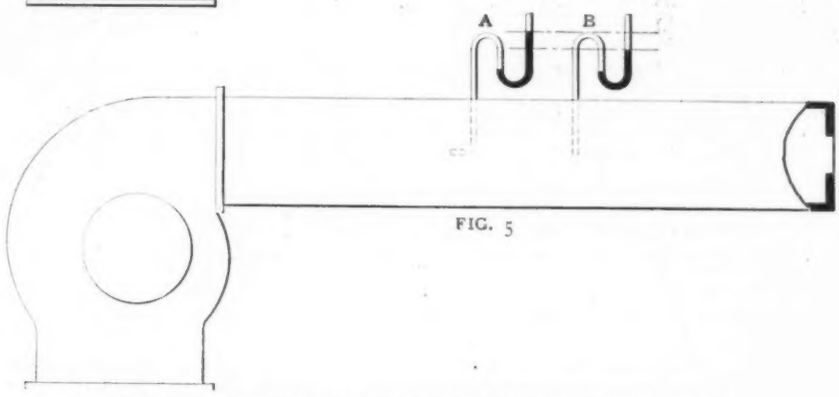


FIG. 5

SHOWING THE EFFECT OF RESTRICTED DISCHARGE OPENING UPON THE STATIC AND DYNAMIC PRESSURES

There are various methods of calculating the capacity and horse-power of ideal fans working against known pressures; but, as these results are greatly changed by variations in the proportions of fan and casing, and also by the resistance imposed by the connecting ducts and flues, it does not seem wise in an article of this kind to take up the purely theoretical side of the subject to any great extent.

The methods given are based upon a series of carefully conducted tests made by one of the leading manufacturers of fans upon their standard steel-plate blowers and exhausters. While the results thus obtained apply particularly to fans of their own make, they may be used with a

fair degree of accuracy for any fan of similar type and proportions.

Before taking up the subject further, let us consider briefly the different forms of pressure involved in the action of a fan.

These can best be explained by the use of a practical illustration. Let Fig. 4 represent a blower with a short pipe attached to the outlet. *A* and *B* are U-tubes, or water gages, for indicating the pressure within the pipe. *A* has the lower end of the tube bent at right angles and turned toward the fan, while the tube leading from *B* is straight. Now, if the fan be revolved with the end of the discharge pipe wide open, the water in the two legs of tube *A* will stand at different levels, thus indicating a pressure within the pipe, while the water in tube *B* remains unchanged. The pressure indicated by *A* is called the *dynamic* pressure and is caused by the air blowing into the end of the tube which is turned toward the fan; or, in other words, it is the pressure due to the movement or momentum of the air. As the end of the discharge pipe is wide open, allowing the air to pass through freely, there is no radial pressure created and therefore none indicated by the tube *B*. Let Fig. 5 represent the same apparatus, but with the end of the air pipe partly closed. Tube *A* in this case shows a higher pressure than before and there is also a pressure indicated by *B*. The latter pressure, which is similar to that in the steam boiler, is called the *static* pressure.

The pressure which produces the flow of air through the pipe, called the *velocity* pressure, is the difference between the dynamic and static pressures.

When the end of the air pipe was wide open, there was no static pressure; hence the velocity pressure and dynamic pressure were equal. But if the pipe should be entirely closed, the pressures indicated by the gages *A* and *B* would be the same, and the velocity pressure would become zero.

When air is allowed to flow from one chamber into another having a lower pressure, there is a definite relation between the difference in pressure and the velocity of flow. This, for air at a temperature of 60 degrees, is given approximately by the formula $v = 66\sqrt{h}$, in which v is the velocity of flow, in feet per second, and h the pressure, expressed in inches of water as indicated by the balanced hight of a column of that liquid in a water gage.

Pressure in inches of water may be reduced to ounces per square inch by multiplying by 0.58, and to pounds per square foot by multiplying by 5.2.

The process may of course be reversed by dividing by the same factors.

In the tests referred to, a pipe the size of the fan outlet and about 12 feet in length was attached directly to the casing. This was provided with a blast gate at the end, so that the area of outlet could be varied at will.

The fan was then run at a constant speed and the dynamic and static pressures measured at a point about midway of the pipe.

Measurements were first made with the outlet wide open and then with ten per cent. reductions until the gate was entirely closed.

The pressure corresponding to the peripheral velocity of the fan-wheel was then determined, by the formula $v = 66\sqrt{h}$, and the various measured and computed pressures plotted as ratios of the peripheral velocity pressure.

Fig. 6 shows two of these curves for outlet openings varying from forty to one hundred per cent., or full opening.

The upper curve shows the ratio between the air velocity pressure and the peripheral velocity pressure $\frac{AVP}{PVP}$ and the lower the ratio between the air velocity pressure and the dynamic pressure $\frac{AVP}{DP}$.

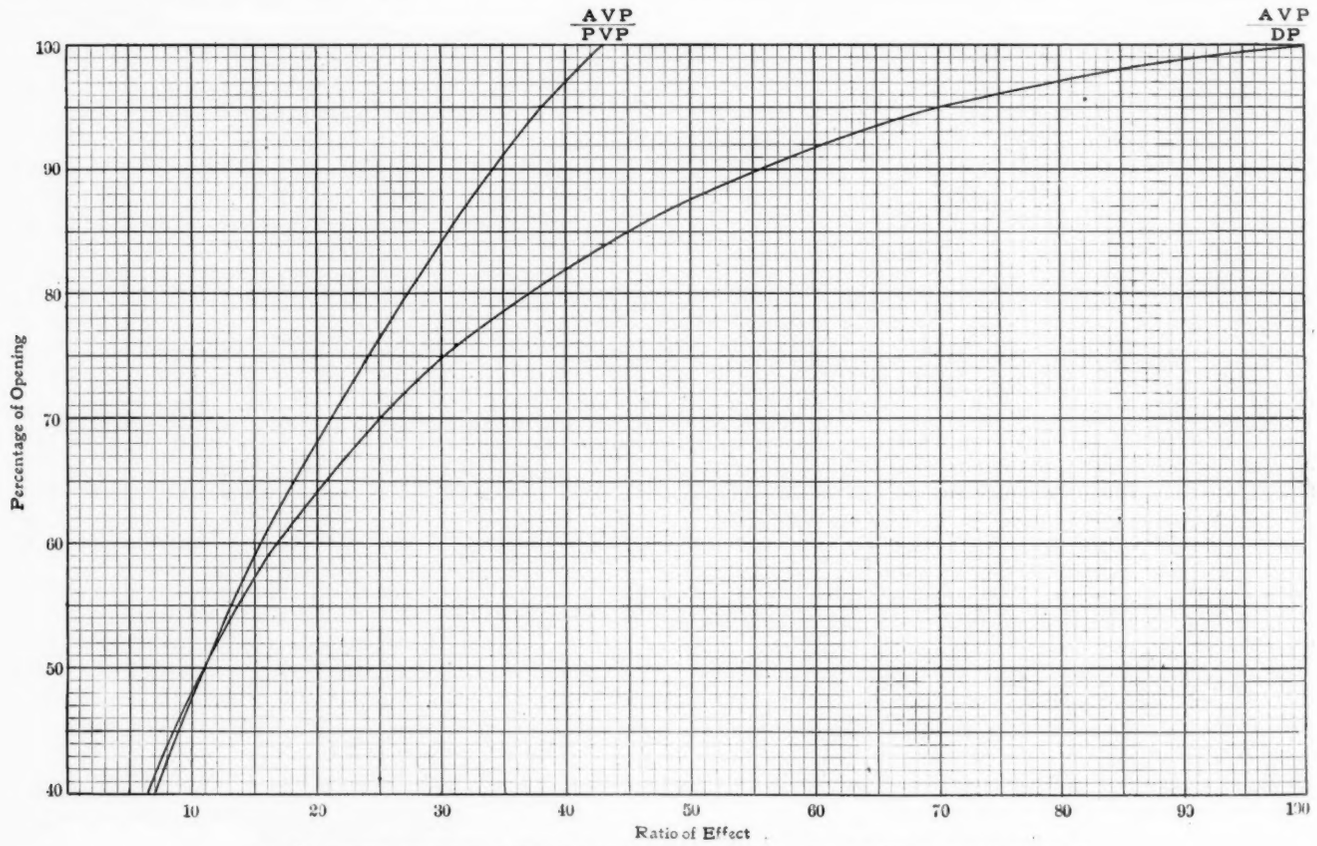


FIG. 6. CURVES SHOWING RATIO OF PRESSURES ATTAINED AND THOSE DUE TO VELOCITY

The practical use of these curves will be shown later.

The capacity of a fan is usually expressed in cubic feet of air delivered per minute, and is equal to the area of the discharge outlet in square feet, multiplied by the velocity of flow.

Referring to Fig. 6 we find the ratio of air velocity pressure to peripheral velocity pressure for a fan with one hundred per cent. opening and discharging into free air, to be 0.43. The relation between the corresponding velocities is equal to the square root of this ratio, or 0.65.

That is, for the conditions stated

above, the velocity of air flow through the fan outlet will be equal to the velocity of the fan tips multiplied by 0.65.

From this it is evident that the capacity of a fan, when discharging into free air, is expressed by the formula

$$Q = 0.65\ C\ R\ O$$

where Q = volume, in cubic feet per minute.
C = circumference of fan-wheel, in feet.
R = revolutions per minute.
and O = area of outlet, in square feet.

Table III gives data relating to the proportions of fans of different diameters, and will be found useful in computing their capacities at various speeds.

The area of outlet is based on the assumption that the width of the wheel is 0.52 times the diameter.

The outlet is made square, with a dimension each way two inches greater than the width of the wheel. When the make of fan is known, the area of outlet given in the catalog may be used instead of the above, with approximate results.

TABLE III.

Diameter of Fan.	Circumference.	Area of Outlet. Sq. Ft.
2' 6"	7.8'	2.1
3'	9.4'	2.9
3' 6"	11.0'	3.9
4'	12.5'	5.0
4' 6"	14.1'	6.2
5'	15.7'	7.6
5' 6"	17.3'	9.1
6'	18.8'	10.7
7'	22.0'	14.4
8'	25.1'	18.6
9'	28.3'	23.4
10'	31.4'	28.7

Example: What volume of air will be delivered per minute by a fan 7 feet in diameter running at a speed of 150 revolutions?

$$Q = 0.65 \times 22 \times 150 \times 14.4 = 30{,}888$$

cubic feet.

This result, it must be remembered, is for a fan discharging into free air; when connected with ventilating ducts or used for mechanical draft a certain correction must be made as described later.

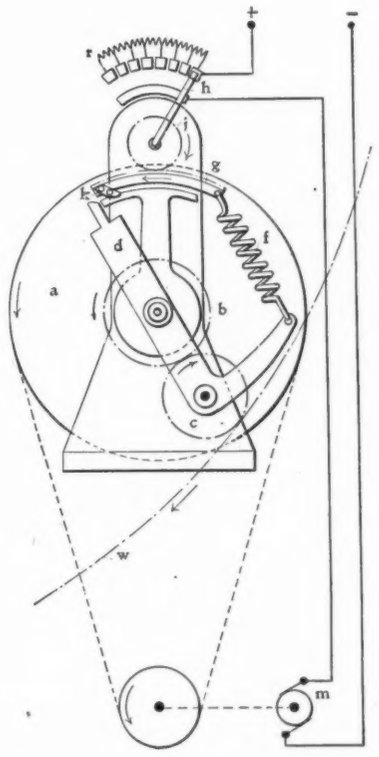
In general, the volume of air delivered by a fan varies directly as the speed, and the power as the cube of the speed; that is, doubling the speed of a fan will double its capacity, but will require $2 \times 2 \times 2 = 8$ times the power. This increase in power applies only to ideal cases, and does not take into account the efficiency of the fan, which, with a free outlet, increases with the speed, up to a peripheral velocity of about 8000 feet per minute. However, the required power increases at a much more rapid rate than the volume of air delivered, even in the case of an actual fan, and this should always be borne in mind when the question of speeding up a fan to increase its capacity is considered.

In cases of this kind, the power required by a larger fan running at practically the same peripheral velocity should be computed and compared with that necessary to run the present fan at the increased speed.

Electric Starting Motor Equipment for Large Gas Engines

By F. E. Junge

It is known that in the majority of cases large gas engines are started by means of compressed air taken from reservoirs which are kept up to pressure by motor-driven compressors. When coupled to a direct-current generator, if a storage battery is available, the dynamo can be run as a motor from the battery and used for starting the engine. For various well understood reasons this mode of starting can be adopted only in rare cases, and it has several drawbacks. The simplest and most reliable way of starting is similar to that applied to large steam engines in Europe: a small electric motor driving through a gear meshing with teeth on the fly-wheel of the gas engine. The one precaution that must be taken is to provide means for throwing the motor pinion out of mesh as soon as the fly-wheel has attained its normal speed.



STARTING DEVICE FOR LARGE GAS ENGINES

The latest device of this type is built by the Felten-Guilleame-Lahmeyer Works in Germany. The motor is thrown in mesh with the fly-wheel and started up by a single turn of the starting wheel, and the gears are thrown out of mesh automatically when the fly-wheel has attained its speed. The accompanying sketch gives a schematic view of the arrangement. The electric motor m drives the disk a by means of a chain. Pivoted on the disk is the toothed gear wheel b, which meshes with another wheel c, supported by the swing lever d in such a way that it can travel around the center of c. The lever d is connected by means of a strong spring f with a toothed segment g, which may be shifted by means of the toothed wheel j and lever h, and which, when moving in the direction indicated by the arrow, catches the lever d and also the toothed wheel c, turning it until the latter meshes with the large toothed gear w of the fly-wheel. Through the electric starter r and lever h the current for the electric motor is switched on at the same time. By turning lever d, and thereby the toothed segment, still farther in the direction indicated, the spiral spring f is more stretched, while the starting resistance is short-circuited in a similar measure. When arriving at its end position, the segment is locked by means of the latch k. As soon as the fly-wheel attains a higher speed than can be imparted to it by the motor, then the toothed wheel c, and therefore the lever d, are shifted in the direction of the fly-wheel travel, thereby unlocking the spring device as shown in the sketch; the toothed segment, accelerated by the spring, moves backward and switches off the current by means of h, thus bringing the motor to rest. The lever h and toothed wheel c are simply pushed aside by the fly-wheel and thrown out of mesh with the gear w.

The attendant has therefore only to gradually turn the hand-wheel on the starting shaft until the lock on the toothed segment comes into operation. While the starting device continues its operation automatically the attendant can go to the main admission valve and open it. This is all that is required.

The above arrangement is built up to tooth pressures of 3000 kilograms and several Continental gas-power plants are thus equipped. For higher pressures the arrangement is preferred in which the motor drives the fixed-tooth wheel by means of a worm gear. In many cases it is desirable to control the operation of starting from a central place, such as the position where the valves of the gas engine are located, in order that starting and admission of the working medium may be regulated at the same time. For this purpose the starting device is equipped with distant control and is actuated by means of a mechanical or electrical relay.

Last year the fatalities in the bituminous coal mines of Pennsylvania were 444 killed inside and 35 outside; that is, one life was lost out of every 345 employees, or 2.90 lives per 1000 employed, or one life for every 249,000 tons of coal produced.

Trials of Engineers in the Early Days—II

Interesting Results of Research in Rural England for Engine Types of the Time of Newcomen and Watt

EDITED BY ALONZO G. COLLINS

"About this time a strolling coal digger came by, who had worked at a pit where was a fire engine, projected by Capt. Thomas Savery, who was reputed as a very ingenious man of science; which he told me 'had no great lever or gear whatever, but did lift from the pit a prodigious river of water,' at which I was much amazed, and considered much if the man might be truthful, as I could conceive of no way by which this might be done, otherwise than as Master Newcomen had devised, but the man being ignorant of all but coal getting, was not able to enlighten me.

"'There was a fierce fire of coals in a fire place, and some coppers, like brewers,' and so little turmoil 'that one in another room need not know ought of what was being done,' and he knew naught else, save there was 'a steelyard on one of the coppers by which to relieve it when the force of the steam was too strong,' of which he told a curious tale of a man, 'who, thinking to make his work sooner done, put the weight on the further end of the steelyard, in order to collect more steam, and this causing the water to go quicker, he then hung a plumber's iron on with the weight, at which the force of the steam not being able to relieve itself, the copper broke into small pieces, with a mighty roar, as of a cannonade, and killed the poor man, and nearly spoiled a score of others.'

"This tale caused me to give much consideration as to the manner in which to contrive a steelyard for my boiler, as I had a notion that such an instrument must be of great advantage, when I bethought me that it could be no more nor less than a device described by the philosopher Papin, and named by him as a safety valve; and I marveled much that I had neglected so obvious a precaution; but I straightway set to, to contrive one, and had it cast at Coalbrookdale; but I still wondered greatly as to the manner by which water might be lifted without a pump, but to no peace of mind, as I could reckon on no way to do it.

"I now had to contrive how to replenish the water in the boiler, for it contrived no great ingenuity to know that with the continual boiling away the boiler must in some certain time be emptied, so as the work of the engine must stop to replenish the boiler, or endanger the spoiling the bottom of the boiler. So I considered whether or no, if a pipe was made from the bottom of the cooling-water cistern to the top of the boiler, the water therein might be of sufficient weight to overcome the force of the steam; but having no knowledge of the manner of reckoning such things, I made a chest of plank, one foot every way, which I balanced on the pit balances, both when empty, and when filled with water, by which I found, after considering the weight of the chest, the water was between sixty-two and sixty-three pounds.

"I then reckoned that the weight of an inch of water one foot tall was closely a half a pound. The distance measured perpendicular from the bottom of the cistern to the top of the boiler, I found was twenty feet, which I reckoned nearly ten pounds of weight, for if one foot was half a pound, two feet must be twice as much, and so on, and this was more weight than was needful, if my reasoning was just, for a force of three pounds of steam had been reckoned for the engine; so a pipe was made as I had measured, with a cock in it to regulate the water.

"The boiler had by this been neatly builded around by brick laid in mortar of sand and clay, as lime mortar is soon spoiled by the fire, with pieces of old cart-tires laid in under the flanch, between it and the brick, the better to preserve them from being so soon burned out.

"Two arches of brick laid in thin lime mortar were laid over the top of the boiler, to preserve the heat, and assist the tightness of the joints.

"To make the piston tight, I plaited some white rope yarns to make a rope that should fit its place, tapering the ends, and of a length to reach around the piston, and the length of the taper more, and beat it into its place, with a plenty of good beef tallow, each ring well beaten, until I had it about four inches deep; I then hammered some bars of iron to a circle to lay on the top of the rope, and contrived some lumps of lead, with handles to lift them with, to lie on the iron bars, so that the rope should stay in its place.

"I had now to contrive to fill the boiler with water, and considered lifting the buckets with ropes to pour them into the cistern for cooling water, whereby the water might find its way by the pipe for replenishing the boiler; but this would have been at the cost of much labor, and I cast about for a better way, which I did by lifting the steelyard of the safety valve and plucking out the poppet, so the water could be poured in, and setting all the laborers in two lines, like infantry, with scores of buckets; the first man did catch up his bucket full from the pool and pass it to his neighbor, and he to his, and so on, to the last man standing astride the boiler top. He poured it into the boiler and passed the empty bucket to the first man in the other line, whereby it was got back to the beginning, until the boiler was full to the top gage-cock.

"For knowing the force of the steam, a tube of copper had been fashioned like a great letter U, one end of which being soldered in the steam pipe, in the other end I poured sufficient mercury whereby to fill it half full, placing thereon a float with a slender spear upright in it, which, coming from the free end of the U-tube, had nicks a half inch apart to show the force of the steam, each nick sufficing for one-half pound of force, as the spear was pushed up with the force of the steam pushing down the mercury in the other end.

"The poppet and steelyard of the safety valve being set back in their places, wood and coals were brought to the furnace, and presently a mighty fire was roaring, and a prodigious cloud of soot was spouting from the chimney; presently the force of the steam began to push up the spear of the steam index, at which, so that all should be in readiness, I set a trusty man at the cock for the cooling water, and then I had like to have died for very shame; for, in my haste to get the engine going, I had forgot to put water in the cistern to serve the engine until the jack pump had been able to supply it.

"However, 'mourning ne'er mended a broken pipkin,' so I cheered up the men to be nimble and pull up water with ropes to fill the cistern; but they were no more than fairly at it than the steelyard of the safety valve suddenly came up with great velocity and, the ball coming off the end with great speed, hit a man in his middle, so that he went down as if dead. The steam issued forth with such a roar that we were like to be deafened—none could hear the other for the turmoil; and with everyone hastening to get as far away as he could, it was as if Bedlam had let loose. As for myself, I was so astounded as to be speechless, not comprehending at first what had caused it; but, quickly recovering myself, I dashed some water onto the fire, by which the heat was moderated, and presently, the uproar having abated, I was

able to take account of our damage, which, beyond the poor fellow who was knocked breathless, was little.

"We had been much hindered in our work by the horses which were used to clear the pit of water, and were often in our way; but I had some little pleasure in that they ever after could not be prevailed on to come near the engine, but snorted and reared in terror at sight of it. And, too, some of the neighboring gentry, who, having knowledge that the engine was to go that day, had gathered from far and near, after our mishap maintained the devil was in it and could never, for a long time, be persuaded to come near it.

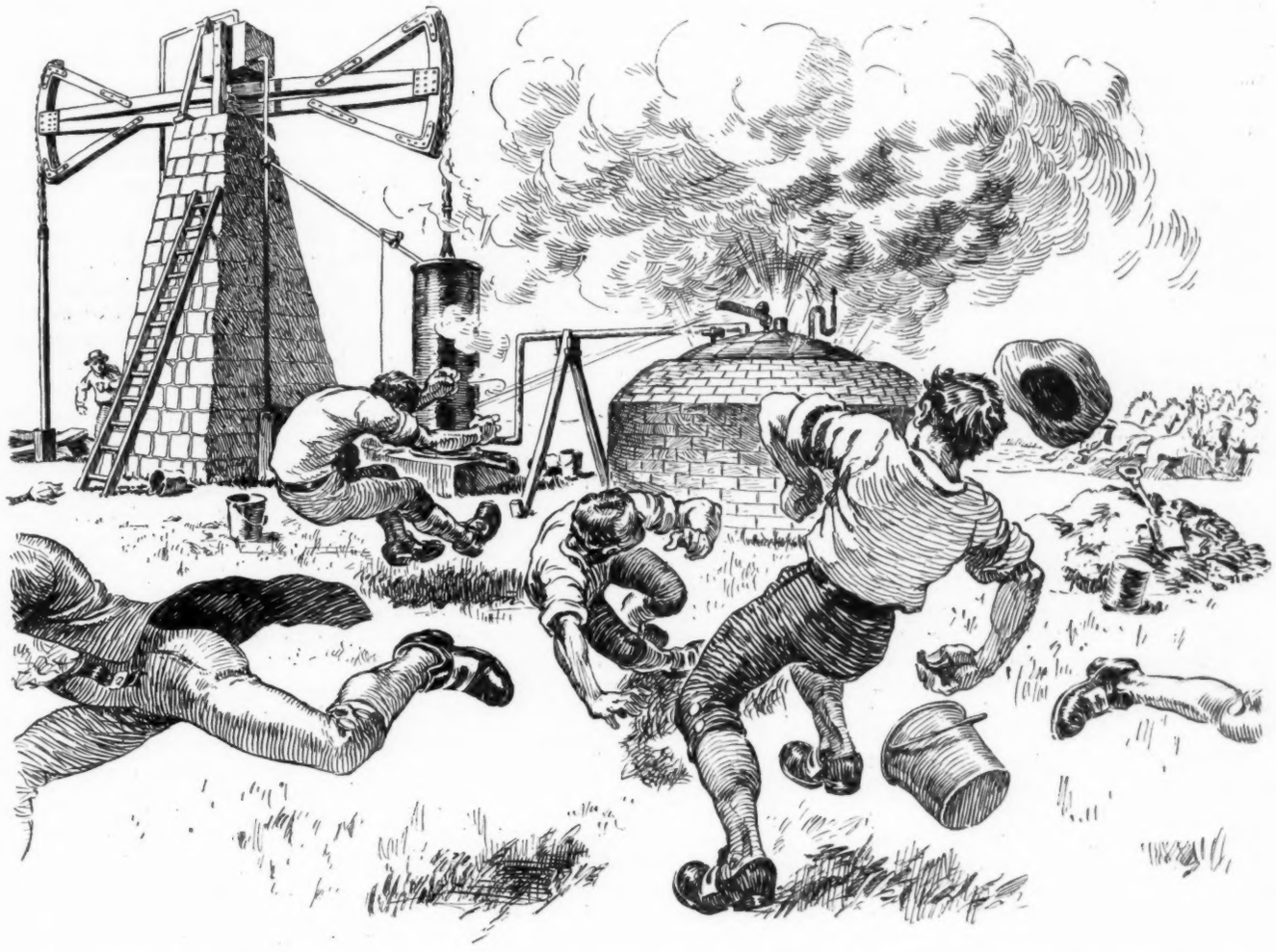
a man at the cooling-water cock, and one to let out the hot water from the case, when it had served its purpose; myself intending to work the steam regulator, which interposes between the boiler and the cylinder.

"The steam now evidencing three pounds of force on the index, I pushed the steam regulator as far as it would go, and the steam rushed into the cylinder with a great snore, which abated the force of the stream in the boiler to that extent as that the steam index sank to its zero nick, but the heat of the fire caused the steam presently to increase its force, so that the index began to rise, and the

trembling of the great chains and lever, there was no motion, the piston declining to descend.

"I then, thinking I had not expelled sufficient air from the cylinder, directed the man to close the cooling-water cock, and the other to open the waste cock, and presently the water case being empty, I opened the steam regulator until the steam came clear from the snifting valve, when I shut the regulator and called for the cooling water, but still to no use, for the piston tarried at the top and would not be coaxed.

"I then called one of the quickest of the lads to carry a rope to throw on the great



"THE STEELYARD OF THE SAFETY VALVE SUDDENLY CAME UP WITH GREAT VELOCITY, AND THE BALL COMING OFF THE END WITH GREAT SPEED HIT A MAN IN HIS MIDDLE."

"The workmen being recovered from their fright I set to and contrived a device for the steelyard, by which to prevent such another mishap, and the boiler being near empty, by reason of the great boiling away while the safety valve was gone, soon as I got it replenished and the cistern filled, I lighted a fresh fire under the boiler, considering meanwhile what new mishap my ignorance might have in store to bring me to grief. But I could see naught more amiss, and the steam beginning to press up the index I again set

snifting valve opened, at which a great flood of scalding water issued forth, followed by a cloud of steam, which presently becoming transparent, I judged the air to be all expelled, then I shut the steam regulator, and called to the man at the cooling-water cock to open it, which he did, and the water case was quickly filled, and water was also overflowing the lip of the cylinder, from that which was on the top of the piston, and the shutting of the snifting valve showed that the vacuum was made; but further than a mighty

lever, as near the cylinder as might be, of which the two ends hung down to the ground on either side, and with the weight of a score of husky men, the piston was persuaded to descend, which it did slowly, as if reluctant, and with much groaning. I then opened the regulator, but now like a contrary horse, the piston refused to ascend, but I cheered up a score of nimble men to grasp the pump-rod with all their might, and so overcame the reluctance of the piston to move, and it was speedily at the top.

"I then shut the regulator and called for the cooling water, at which the piston came down a little way and stopped, hesitating; descended a little more, when the men jumped to the ropes, and willy-nilly, down it came. I then perceived the fault to lie in the piston being too tight in the cylinder, but considered that if it could be forced to go a few strokes, it might relieve itself. I called for the waste cock to be opened, and when the case was empty, I snifted the water from the cylinder as before and the piston made the ascent without the help of the men, and when the vacuum was made, made its descent with less help from the men; and so by cheering the men to be nimble, we at last had persuaded the engine to be tractable and had presently come to be so nimble in working the cocks and regulator, that the great lever, with an infinite din and jangle, was rising and falling with great regularity.

"A mighty stream of water was issuing from the mouth of the pump, and the hostlers, having nought to do, with many sour looks at the engine and myself were leading the horses to their stalls."

Reynolds straightened up with a sigh, perspiring from every pore. He had in imagination been in the old engineer's shoes, sharing his labors and anxieties. He said aloud: "I don't know who you are, but I would like to shake hands with you; you sure were a crack-a-jack, if you know what that means. Here have I fairly raised the roof off the plant, because some lunkhead has made a mistake in the plans, and the piping won't fit, and ripped the shop men 'up the back,' because a cylinder was a thickness of tissue paper out of true, and yet you, poor devil, were tickled to death because you could only just enter the end of your little finger between the piston and cylinder at the worst place.

"I wonder if I would have done as well as you, with never the scratch of a pencil in plans, and two-thirds of the stuff to build on the grounds by yourself. I'll bet you rebuilt that engine though, before you left it."

The next morning the reading of the journal was resumed as follows:

"I had now to train some nimble lad to work the engine, as I had no notion to be tied to it, and besides someone must know to work it at night, as the pit had such a store of water rushing in that the pump must not stop for long; and in any case the labour being no more than what a lad's strength can perform for a day together, and as easily learned as the driving of a horse in a tub-gin.

"So I picked on a likely youth, Potter by name, who had been of some little use in the work, and by reason of him contriving always that his task should be done with as little use of his own strength as might be, I reckoned that he might in time come to be of use as a trusty workman. When he came to be nimble in the working of the steam regulator I trained cords from the cooling-water cock, and the waste-water cock, to his hand, so he alone, without need of stirring from his place, could make the engine go; and a man to carry coals, with a trusty man for fireman, who should look over all, and control the others, contrived to do all that had heretofore employed more than fifty horses to do.

"Sometime before nightfall I had set another youth with young Potter to be trained in working the engine, and a fireman, for the night shift, and contented myself with idle contriving in my mind of divers betterments to the engine.

"I was pleased that the device for the replenishment of the boiler did do its work so well; but it was not altogether to my liking, as, if the water was too much boiled away, by the heedlessness of the fireman, the injection of so great a quantity of cold water quickly abated the force of the steam, so as the work of the engine must stop until its force had been augmented; and I considered whether I might contrive some way to warm the water on its way to the boiler, but could discover no way whereby I might compass it.

"I then considered if I might discover some way so as by the abatement of the water in the boiler, by the boiling away, the cock might be opened, and by the augmenting of the water in the boiler the cock might be shut; and this I accomplished by contriving a little cistern over above the boiler, a foot less in heighth than the great cistern, from which it should derive its water, with a buoy-cock to regulate it, and keep it always full, and a pipe straight and true, from the bottom of the little cistern to the top of the boiler; with a float contrived to sit on the water in the boiler, with a spear set upright on which was a plug to match the pipe, so when the water was sufficient in the boiler, the plug stopped the pipe and no water might enter the boiler; but if the float sank ever so little, by reason of the water boiling away, then the plug coming from the mouth of the pipe, the boiler was replenished and so efficiently did its work that, whereas before, owing to the abatement of the force of the steam by reason of the great quantity of cold water let in to replenish the boiler, the engine worked but eight strokes in a minute; afterward it worked ten strokes in a minute; yet I considered that if the water could be made warm it would work to greater advantage.

"There was, however, but scant time for idleness; the engine was often gone wrong, our greatest task being to pack the piston, the rope being so soon spoiled; and one night I was called to in my sleep that the rope was gone and no yarns to make new ones; but I plucked off my breeches and the fireman his, and with them so packed the piston that the engine went until new yarns were brought; happily it was night, or we had made a strange sight to folk who came to look at the engine. I made bold to seek the owners of the pit for new breeches, and they gave us them, but warned us they would do so no more, as they had no notion to trade old clothes for new, we must look to it that we had yarns."

After Grafters in England

The new Prevention of Corruption Act, 1906, has now come into force, and severe penalties await the givers and receivers of secret commissions who are detected in their dishonest dealings. We are indebted to the secretary of the newly formed Secret Commissions and Bribery Prevention League for a preliminary circular giving details of the purposes and program of this body, and also the terms of the act which it is intended to support. From the latter we gather that any person corruptly giving or receiving a secret commission, or attempting so to do, is liable on conviction to a term of imprisonment not exceeding two years, with or without hard labor, or to a fine not exceeding £500, or to both fine and imprisonment. This fact, when it becomes known, will, we hope, have a wholesome effect on those who have been dishonestly filling their own pockets at the expense of other people. One large employer of labor told us the other day that he had notified his managers, foremen and other responsible employees of the terms of the act, and that he had also communicated with all the firms from whom he bought goods. To the latter he had promised that no accounts would be changed for at least six months, so that the cessation of gratuities or commissions to employees should not prevent the goods of all the firms concerned having a fair trial on their merits. We wish the new league every success in its efforts to suppress this detestable practice of secret-commission giving.—*Mechanical Engineer.*

Engine-stops and Fly-wheel Insurance

Following upon our editorial under the above title in the February issue, we are informed that the Casualty Company of America has decided to make a reduction of twenty per cent. from the regular rates, not only upon fly-wheel, but upon employers' liability insurance, in all cases where the Monarch engine-stop and speed-limit system is installed, for the same reason and upon the same principle that a manufacturer receives a reduction in his fire insurance rate when a sprinkler system is installed.

Elementary Electrical Engineering—XVI

Electrical, Magnetic and Mechanical Losses in a Motor; Computing Motor Efficiency and Percentage of Each Loss

BY CECIL P. POOLE

When a motor is in operation, the power represented by the watts taken from the supply circuit is divided into five parts: First and largest, the external work done by the motor; next largest (usually), the loss due to the electrical resistance of the motor circuits; the loss due to hysteresis in the armature core; that due to eddy currents in the armature core and field-magnet pole-faces, and that due to mechanical friction. The cause and effects of hysteresis and eddy currents were explained in the January article of this series.

The Shunt-wound Motor

The loss in any electrical circuit due to resistance is equal to the product of the resistance of the circuit and the square of the current in it, or the product of the current and the voltage required to force it through. Thus, if the motor be of the simple shunt-wound variety and the resistance of the field winding be 800 ohms, the voltage being 110, the current in the field winding will be $110 \div 800 = 1.375$ amperes, and the loss will be $110 \times 1.375 = 151.25$ watts. Now if the resistance of the armature circuit (the winding, commutator connections, brushes, brush-holders and connections) be 0.171 ohm and the full-load armature current 38 amperes, the loss in the armature circuit will be $38^2 \times 0.171 = 247$ watts.

The losses due to hysteresis, eddy currents and mechanical friction are difficult to separate, but the sum of them is easy to determine by elimination, or by simple tests. If the armature current at which the motor delivers its rated output is known accurately, the core and friction losses can be ascertained by elimination. In the case already cited, the full-load armature current was 38 amperes, the machine was a 5-horse-power motor, the circuit voltage 110. In order to deliver 5 horse-power at the pulley, 3730 of the watts taken in by the armature must be devoted to net work; the total armature watts were $38 \times 110 = 4180$, and the loss in the armature circuit due to resistance was 247 watts. The resistance loss plus the net work add up $247 + 3730 = 3977$ watts, so that the core and friction losses must have been $4180 - 3977 = 203$ watts.

With the machine in operation at full load, the hysteresis, eddy currents and friction exert a backward drag or braking effect on the armature; the electrical resistance of the circuit does not. The sum of the net load and the losses due to hysteresis, eddy currents and friction, expressed in watts, is equal to the product of the armature current and counter e.m.f. In other words, the resistance of the armature circuit causes a drop in voltage, the counter e.m.f. of the armature making up the difference between this drop and the impressed voltage; most of the counter e.m.f., multiplied by the armature current, represents the net external work done while the remainder, multiplied by the current, represents work done in overcoming the backward drag of friction, hysteresis and eddy currents. In the case just cited the armature-circuit resistance was 0.171 ohm and the full-load current 38 amperes; the drop in the armature circuit, therefore, was $0.171 \times 38 = 6.5$ volts, and the counter e.m.f. was $110 - 6.5 = 103.5$ volts. Of this, 98.158 volts, with the armature current, were devoted to doing the net external work of 5 horse-power (5 horse-power $= 5 \times 746 = 3730$ watts, and 98.158 volts $\times$ 38 amperes $= 3730$ watts) and the remaining 5.342 volts, with the armature current, did the work of overcoming the drag due to iron losses and friction: $5.342 \times 38 = 203$ watts, which agrees with the previous calculation.

If one does not know what the full-load current is, accurately, the only way to determine it is to make a brake test of the machine, either driving a dynamo the efficiency of which is known at all loads or running with some form of brake dynamometer applied to the armature shaft. There is also a method of testing out the core and friction losses of a machine by driving a similar machine as a dynamo, but it is rather too complex for discussion here.

Having ascertained the armature losses and added all of them to the loss by resistance in the field winding, the efficiency of the machine may be computed by means of the formula:

$$\frac{100 \times \textit{Net power}}{\textit{Net power} + \textit{Losses}} = \textit{Efficiency},$$

the efficiency being expressed as a percentage. Thus, in the case of the small motor just cited, the loss by resistance in the field winding was 151.25 watts, that in the armature circuit was 247 watts, making a total resistance loss of 398.25 watts; the losses due to hysteresis, eddy currents and friction aggregated 203 watts, so that the total losses were $398.25 + 203 = 601.25$ watts. The net output of the machine was 5 horse-power, which is equivalent to 3730 watts. The efficiency, therefore, at full load, was

$$\frac{100 \times 3730}{3730 + 601.25} = 86.118$$

per cent., or practically 86.12 per cent.

It is usually desired to know just what the percentage of loss is in each part of the machine. This is computed by the formula:

$$\frac{100 \times \textit{Individual loss}}{\textit{Net power} + \textit{Total losses}} = \textit{Per cent. individual loss.}$$

Thus, the loss in the field winding was 151.25 watts; the percentage of loss in that part was

$$\frac{100 \times 151.25}{4331.25} = 3.492 \text{ per cent.}$$

The loss in armature-circuit resistance was 247 watts, or

$$\frac{100 \times 247}{4331.25} = 5.703 \text{ per cent.},$$

and the loss by hysteresis, eddy currents and friction combined was 203 watts, or

$$\frac{100 \times 203}{4331.25} = 4.687 \text{ per cent.}$$

It will be found upon adding up these individual percentages and subtracting their sum from 100 that the remainder is the efficiency of the motor, as determined by the first calculation. Thus, the sum of the individual percentages of loss is $3.492 + 5.703 + 4.687 = 13.882$; subtracting this from 100 leaves 86.118, the result of the first calculation.

It is difficult to separate the losses by hysteresis, eddy currents and friction, as stated before, but it can be done. The method will be explained fully in a subsequent article devoted to the testing of dynamos and motors.

The Series Motor

The case of the series-wound motor is very similar. The losses are, of course, due to the same causes as in the shunt-wound machine and the efficiency calculation is the same. It is scarcely necessary to discuss the case in detail; an example will suffice to illustrate what small differences there are between this and the shunt-wound motor. A 5-horse-power series-wound motor, operating on a 110-volt circuit, embodied the following data:

	Ohms resist.	Volts drop.	Watts lost.
Field winding	0.1	3.9	152.1
Armature circuit	0.132	5.15	200.8
Hysteresis, eddy currents and friction	.....	....	207.1
Total resistance drop	.....	9.05	.....

The total drop due to resistance being

9.05 volts, the counter e.m.f. of the armature was $110 - 9.05 = 100.95$ volts. Since the loss due to hysteresis, eddy currents and friction was 207.1 watts, the amount of counter e.m.f. absorbed by this combined loss was $207.1 \div 39 = 5.31$ volts, leaving 95.64 volts of counter e.m.f. available for net external work. Multiplying this by the current gives $95.64 \times 39 = 3730$ watts as the power devoted to external work, and this is the equivalent of 5 horse-power.

The total losses add up $152.1 + 200.8 + 207.1 = 560$ watts, so that the efficiency of the machine was

$$\frac{100 \times 3730}{3730 + 560} = 86.95 \text{ per cent.}$$

The loss in the field winding was

$$\frac{100 \times 152.1}{4290} = 3.54 \text{ per cent.}$$

The loss in the armature winding was

$$\frac{100 \times 200.8}{4290} = 4.68 \text{ per cent.}$$

The other losses were

$$\frac{100 \times 207.1}{4290} = 4.83 \text{ per cent.}$$

The total losses aggregated, therefore, $3.54 + 4.68 + 4.83 = 13.05$ per cent., leaving 86.95 per cent. for useful work, as shown by the efficiency calculation.

To join two pieces of either iron or steel together by brazing, the ends to be brazed should be thoroughly cleansed of grease and acid, and after applying a solution of borax to each, the joint should be heated in a clear charcoal fire, or gas flame, and borax spelter sprinkled on the joint after it is thoroughly heated. When the spelter melts and runs along the joint, withdraw the metal from the fire and allow it to cool slowly, after which the joint may be finished as smoothly as desired. The joint before brazing should be clamped by some means to hold it in place.

Rigging up to Turn and Refit Large Pistons—A Crank-pin Turning Device—etc.

By Dixie

A number of articles have appeared from time to time, on doing large jobs on small machines, but I have yet to come across one in the same class with that which I here describe. This not only is a large job done in a satisfactory manner in a shop having small tools only, but also it shows how an efficient special machine was built from material rescued from my old friend and standby, the scrap heap.

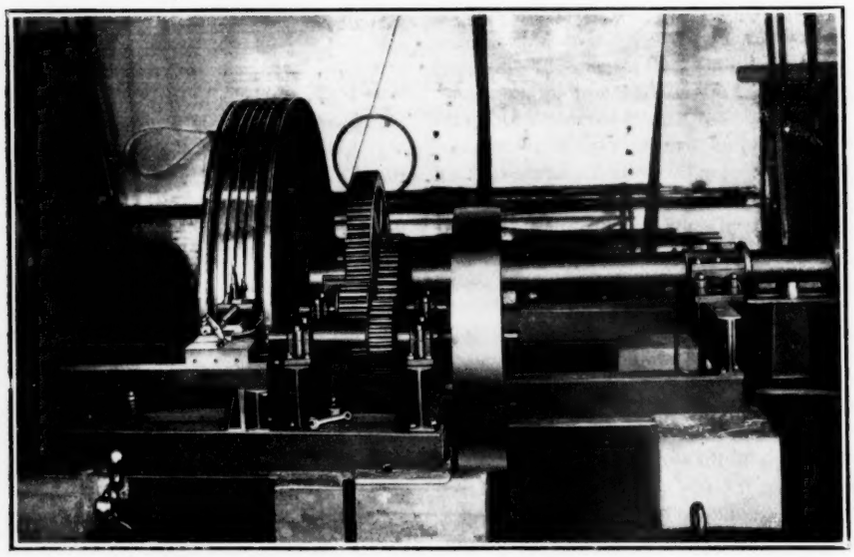


FIG. 1. A LATHE BUILT FROM SCRAP

The plant where the work was done is a large power station with eight compound engines having cylinders 42 and 86 inches by 60 inches stroke; steam pressure, 180 pounds; revolutions per minute, 75. These engines were built to develop 4500 horse-power, but are run anywhere up to 6000 horse-power. Six engines are in commission at a time while a standby engine is kept just turning over so that at a moment's notice, in case of accident, it can be started at full speed. The remaining engine is undergoing any necessary overhauling.

This job has in the past been a man-killer, as far as chief engineers are concerned, but the present incumbent, A. W. Jurisch, seems to be pretty far "ahead of the game" at present and likely to remain so.

The pistons of the low-pressure cylinders are 86 inches diameter by 15 inches deep. Two inches from each face there is a packing-ring groove *B*, Fig. 2, 1¼ inches wide. The pistons cost when new $664, and when put in are about 0.032 smaller in diameter than the cylinders. After a year and a half they are found to have worn approximately ⅜ inch, and if they could not be repaired would be a total loss.

Mr. Jurisch decided to try cutting grooves in one of the pistons and lining it with babbitt. This was done and gave such highly satisfactory results that as they become worn and can be spared the other pistons receive the same treatment.

The largest lathe in the repair shop is a 24-inch Putnam with a bed about 10 feet long. It was, of course, out of the question to turn up the 86-inch pistons on this lathe, but it came in for the work on the piston-rods later on. Out in the scrap heap was a bent piston-rod—not very much bent, but just enough to put it out of business for the work for which it was intended.

This was brought in and a couple of journals turned on it about six feet apart and true with the taper seat for the piston. Two bearings were fitted to these journals. Masonry piers were built 18x24 inches and high enough to suit the job. To these 10x5-inch I-beam were secured by bolts built into the masonry. On top of the 10x5-inch I-beams, 1-as inch I-

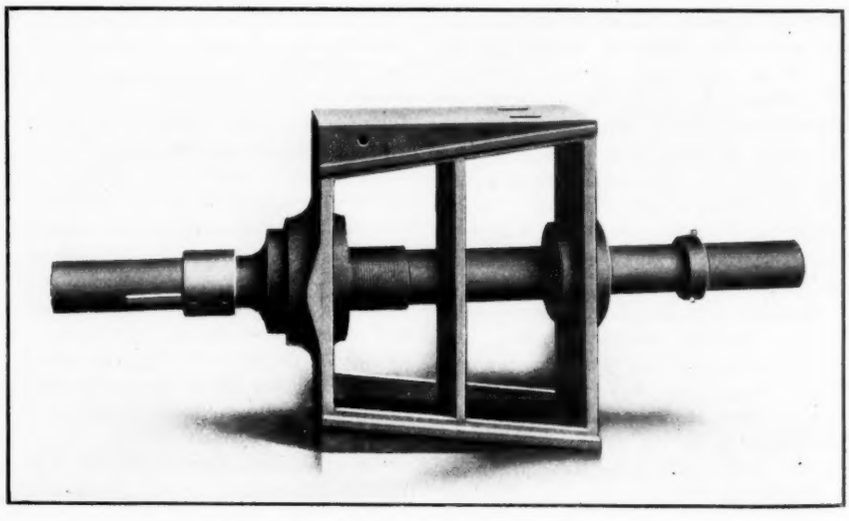


FIG. 3. RIG FOR TURNING CROSSHEAD SHOES

beams were bolted, and to these the two bearings were secured, all of which is shown in the half-tone, Fig. 1. The slide at the left, upon which the tool slide is mounted, is equipped with a carriage having screw feed, which serves for longitudinal feed for the tool. The tool slide is taken from the 24-inch lathe previously referred to. The pistons are mounted on the end of the piston-rod which forms the spindle of the machine, and six dovetailed

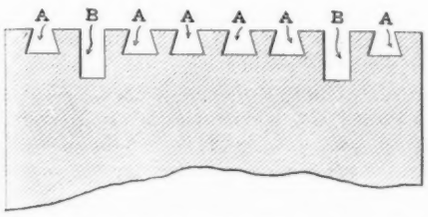


FIG. 2. SECTION OF PERIPHERY OF PISTON

grooves 1¼ inches wide by ¼ inch deep are turned in them, as shown at *A*, Fig. 2. This job took eight hours, which is pretty quick time. After grooving, a wooden form lined with asbestos board and taking in about 1/6 of the circumference is clamped to the piston and the grooves are poured with babbitt. The six segments of babbitt are not allowed to abut, there being a space of about ½ to ¾ inch between their ends. After all the segments are filled with babbitt, the babbitt is pened into the groove, the pening at the same time that it compresses the metal spreads it so that the segments creep toward each other, making a perfectly tight joint at their ends. After the segments are all pened solid they are turned to size, the whole job taking but three days.

The first cylinder fitted with one of these doctored pistons has now a fine glazed surface on it, and is the only one that has.

The crosshead shoes of these engines are also babbitt lined and are turned to size in the same fixture, but between the bearings. The rig shown in Fig. 3 shows a spare crosshead secured to a shaft with journals on it which fit the bearings on the turning rig. The large gear on the spindle of the turning rig is split so that it can be easily shifted from the one job to the other. The tool slide is also portable, of course.

A Crank-pin Turning Rig

Fig. 4 shows a crank-pin turning device devised by Mr. Jurisch. In this rig *A* is a rotating sleeve journaled in the casting *B*, which slides on the angle casting *C*, being actuated by the screw *D*, working in the nut *E*. A worm *F* meshes with worm teeth cut in the center of the periphery of *A*. The worm *F* is driven through the worm gear *G*, the worm *H*, and the shaft *J* by the pulley *K*. The sleeve *A* carries the turning tool *I*. The sleeve *A* is not shown in halves in the illustration, but can be so made. It is, of course, understood that no attempt has been made to make the drawing to scale.

The face *X* of the angle casting *C* is set true with the crank web, and the sleeve *A* concentric with the crank-pin. The tool *I* is set to depth of cut and the machine is started. The feed is by means of a star wheel on the end of the screw *D*.

Turning Long Piston-rods on a Short Lathe

The spare piston-rods for these engines are about two feet longer than the longest lathe in the shop. The threads on either end for the nuts and the taper for the pistons have been left a "little full." When it came to fitting a new rod, these oversizes were noticed, and it was up to Mr. Jurisch to devise some means for doing the job of turning them to size. This is shown at Fig. 5. *A* is the piston-rod, *B* is a small cast-iron piece drilled for three cap screws. The ends of the rods are drilled and tapped so that the piece *B* can be secured to it. The piece *B* is caught in the chuck and the rod is supported in a steady rest having wooden jaws. Even if the piece *B* is not central with the rod, the chuck jaws can be easily set so that the rod runs true. This made it possible to reduce the threads and the tapers, instead of re-threading the nuts,

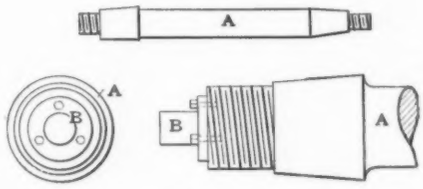


FIG. 5. RIG FOR TURNING LONG RODS

which would, of course, prevent their being interchangeable, as the threaded portions on the spare rods were not all exactly alike.

A Piston-nut Wrench

Another useful tool is shown at Fig. 6. It is used for tightening the piston nuts when in place in the engine. The flange *A* fits, with a little play, the manhole in the cylinder cover. The piston is brought to top center, the wrench is put in place, and the flange *A* steadies it so that there is no danger of its being displaced while the wrench bar is turning it.

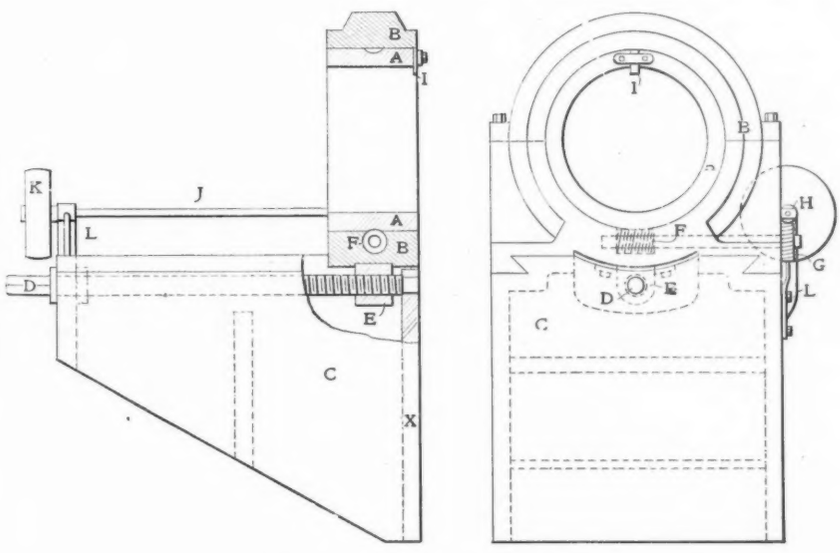


FIG. 4. CRANK-PIN TURNING RIG

The *Engineering Journal* of Canada says: The power efficiency of certain soft coals when used in the gas-producer plant is two and one-half times greater than when used in an ordinary steam-boiler plant. This is shown by geological survey tests, which also develop that the softest grades of bituminous coal and lignite are equal in power production with the best grades, even surpassing anthracite for some purposes.

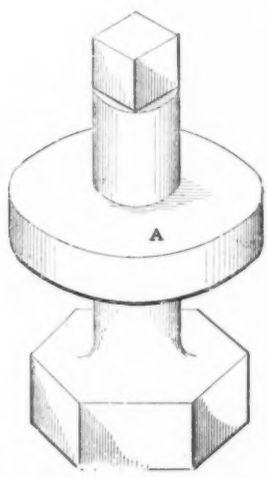


FIG. 6. WRENCH WITH STEADY FOR PISTON NUTS

Troubles Caused by Faulty Piping Work

Examples of Mistakes Often Made by Engineers and Fitters, and Suggestions How to Remedy or Avoid Them

BY H. A. JAHNKE

In certain steam plants of which I have knowledge, slight attention is given to the layout of steam and water piping, especially with regard to the proper appearance and the most convenient arrangement of valves and other fittings. In many cases more valves, ells, tees and unions are used than is necessary, and it very often happens that some part of the power-plant installation does not work right because of the improper arrangement of piping.

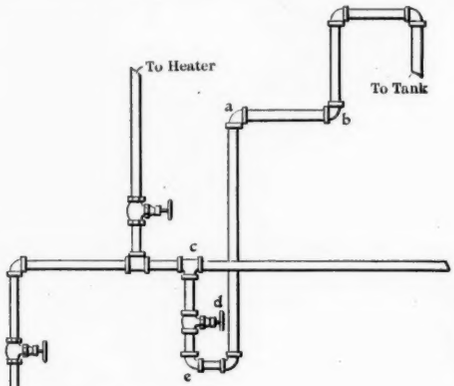


FIG. 1

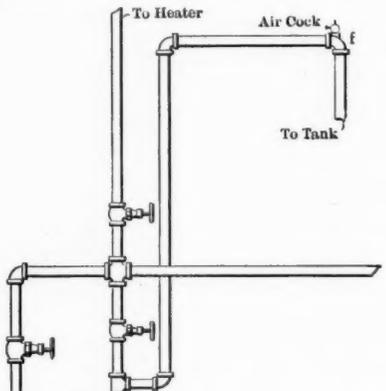


FIG. 2

Fig. 1 shows the original layout of the supply pipe from the city water main in our plant, which furnishes water for various purposes. It was discovered that there were more ells in this job than necessary, so the layout was changed to that in Fig. 2, doing away with the two ells *a*, *b*, Fig. 1, and the tee *c*, thereby making a much better looking job. A section of the pipe between *a* and the valve *d* used to freeze during cold weather, as there was no way of draining it, so when the layout was changed provision was made for draining the pipe by inserting a tee in place of the ell *e*, and connecting a drain-pipe by means of a short nipple and a valve; an air-cock *f*, Fig. 2, was located in the ell at the tank to admit air in the pipe when the drain valve was opened.

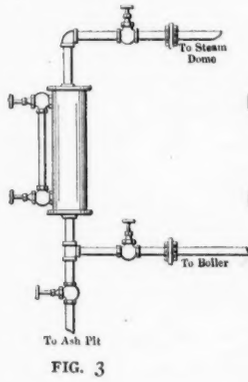


FIG. 3

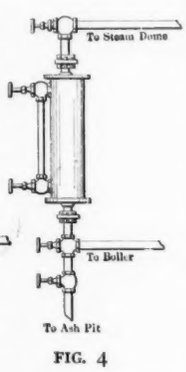


FIG. 4

A few years ago I had charge of a boiler in which the pipes on the water column were arranged as in Fig. 3. There was no way of cleaning the lower pipe from scale which might have lodged there. I changed this arrangement to that shown in Fig. 4, which was not only more attractive, but provided means for cleaning the pipe by removing the bonnet from the cross-valve and running a rod through the pipe. It will also be seen in Fig. 3 that the union is between the boiler and globe valve. When this union needed a new gasket there was no way of shutting off the pressure until the boiler was washed out.

In Fig. 4 the location of this union was changed to between the valve and water column, after which the union could be repacked at any time by closing the valve shutting off the pressure for the boiler. The pipe between the boiler and the cross-valve was made shorter, which brought the column closer to the boiler front. In the steam pipe from the water column to the steam dome an angle-valve was placed, instead of an ell, and in this connection also the union was re-located between the valve and water column.

Fig. 5 depicts an excellent plan, that of locating the two water columns on twin boilers close together; that is, in the center of the boiler fronts. Of course this can only be done where the boilers are set side by side. In this arrangement the pipe can be cleaned when the boilers are washed out, by removing the plugs *a a* and *b b*, and the bonnet on the angle-valve. It is good practice to cover the threads on the plugs and the bonnets on the angle-valves with graphite mixed with oil, so the plugs and bonnets will come out easier the next time.

Fig. 6 illustrates a 1¼-inch steam pipe which was used to supply the boiler feed-pump, heat water in a large tank, furnish steam for the whistle and for various other purposes. It was the whistle which gave the trouble, as it could be barely heard when blowing, due to not getting enough steam. As there were three ells in the pipe leading up to the whistle, besides all the ells, tees and valves in the other sections, when the pump was running and steam was being used for the tank, pump and tank taking a 1-inch pipe each, there was not much steam left for the whistle. The pipe to the whistle was changed, as shown in Fig. 7, and connected to the main header, 45-degree ells being used in place of the three ells and other fittings in the old arrangement. Af-

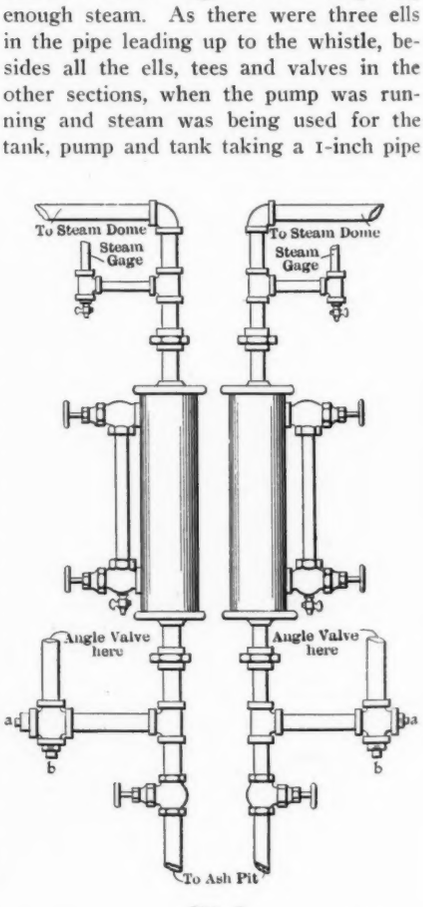


FIG. 5

ter this improvement the whistle gave a loud sound.

It will also be seen by reference to Fig. 6 that there are two valves in the pipe which furnishes steam for the boiler feed-pump. It is not necessary to have two valves in this pipe, because if the valve at the pump gets out of order the valve at the boiler can be closed while the pump valve is repaired.

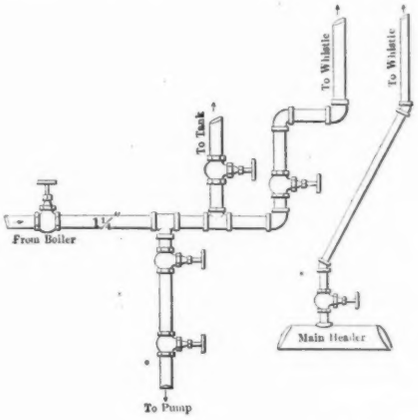


FIG. 6 FIG. 7

I have often noticed that in some steam plants when changes are to be made in the piping the owner generally wants to use "any old piping" which may be lying in stock, the result being that bushings and reducers are used which not only make an untidy job, but frequently the arrangement does not work as it should. When making changes in any piping system, pipes of the right size should be used all the way through, whether in stock or not, as it pays in the long run and besides saves trouble and worry for the men in charge.

This subject calls to mind an experience I had with the owner of a steam plant. The building was lengthened eighty feet and steam pipes were wanted for this extension. The owner gave me orders to put in the pipe as shown in Fig. 8. The old building is about eighty feet long and contains 640 feet of 1¼-inch pipe. The same amount of piping was to be used in the new building, also. (In the sketch only half of the piping is shown.) Exhaust steam is used for heating. The owner wanted to put a short two-inch nipple in the branch tee of the piping in the old building, the branch tee for the new building to be placed on top of this. Just imagine, through this two-inch nipple enough steam must pass to supply the 640 feet of 1¼-inch pipe in the new building. It probably would have worked if live steam were used. The reason why the owner wanted the job done this way was because he thought there would not be pitch enough in the pipes to drain them if they were run in a straight line through the two buildings.

Upon inspecting the pipes in the old building I found that they had a 1¼-inch pitch to the foot, and in order to run the pipes from the old building to the new, I raised the pipes in the old building so that they had only 1/16 of an inch pitch in every foot. Then I disconnected the branch tee from the old pipes, put the coupling on and continued the pipes up to the end of the new building. See Fig. 9. This did away with two branch tees the owner wanted used, besides making a neater job; and there was plenty of pitch to the pipes, so that they would drain.

A mistake which some engineers and steam fitters make is in placing a flange or union on the pressure side of a pipe, between the boiler and the valve. When the gaskets in these joints blow out or leak, there is no way of renewing them until the pressure is out of the boiler. Whenever possible a flange or union should be placed between a valve and the engine or pump. In cases where this is not possible, a good grade of packing should be used in the joint, and the work done by an experienced man.

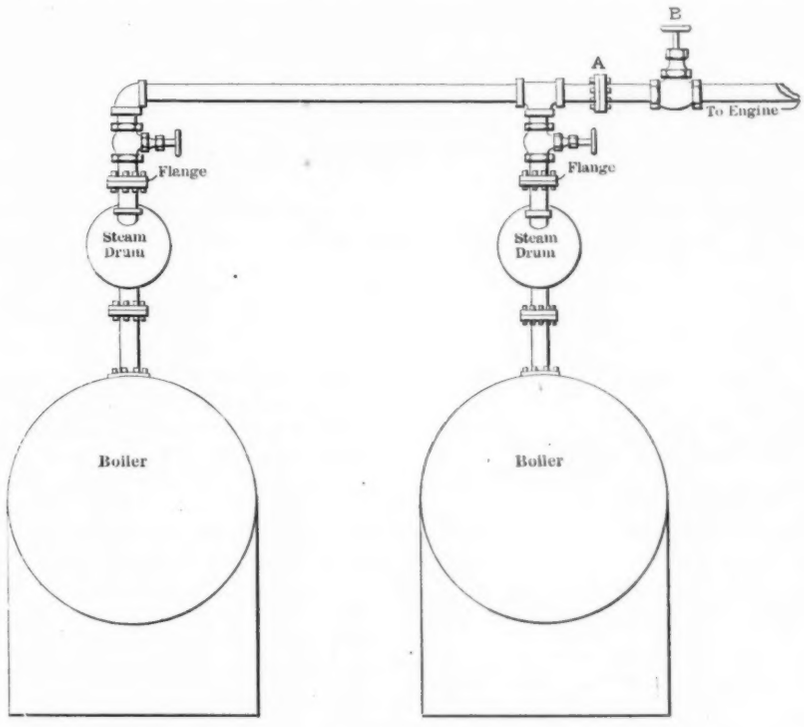


FIG. 10

Fig. 10 illustrates the steam drums and main pipes on two return-tubular boilers. It will be noticed that the flanges are located between the stop-valve and the boiler. The proper place for them is between the valves, ell and tee, so that if the gasket should blow out or leak, the valves could be closed and repairs made without letting the steam out of the boiler. It will also be seen that the flange *A* is on the pressure side, while it should have between the valve *B* and the engine valve, for then if anything goes wrong with the flange or the valve at the engine, the valve *B* can be closed while repairs are made. The flange *A* could have been easily located in the right place at the start, without more expense or extra labor.

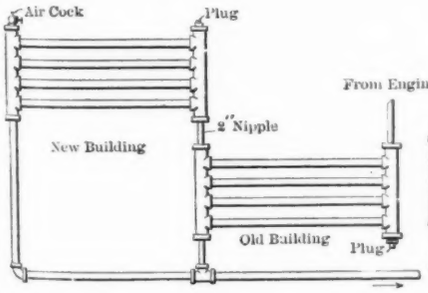


FIG. 8

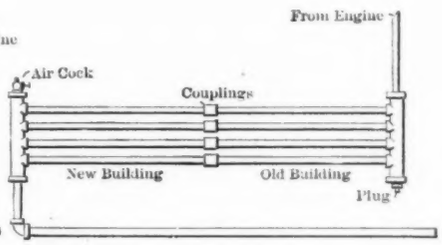


FIG. 9

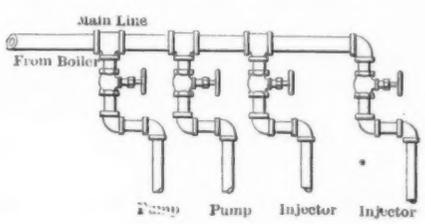


FIG. 11

Another error consists in running an independent line of pipe to each pump, injector or other apparatus, direct from the steam dome or main steam header. A better plan is to run a main line with branches to each pump or injector, having a valve near the main line on each branch, and a valve near each pump or injector. The valve in each branch near the main line, as shown in Fig. 11, is for the purpose of shutting off the steam in the branch to prevent water hammer when the pump is laid up for repairs, and the other pumps are running.

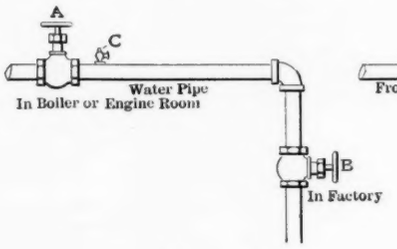


FIG. 12

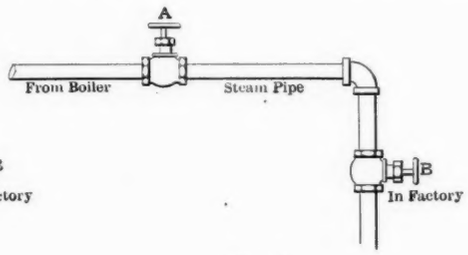


FIG. 13

When piping a steam plant in which the pipes which run from the boilers to the factory are liable to freeze during the night, when the plant is not running, the steam and water pipes should be arranged, so they can be drained, something on the order shown in Fig. 12. At night the valve *A* on the water pipe in the boiler room or engine room is closed, the valve *B* in the factory is opened, the air-cock *C* is opened to admit air, and this will cause the water to drain out of the pipe.

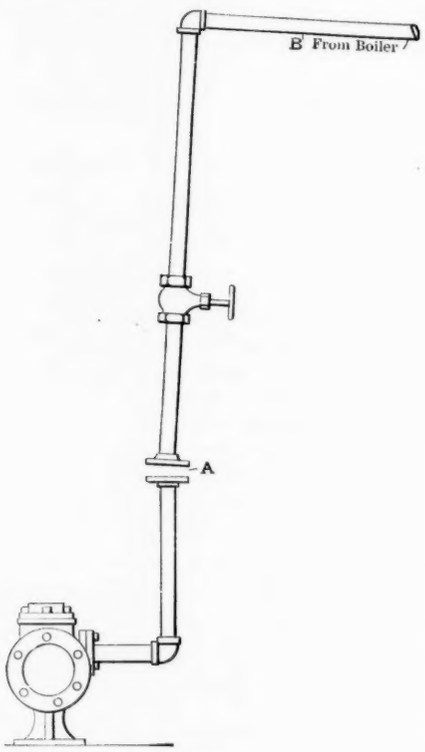


FIG. 14

Steam pipes which run from the boiler into the factory can be drained by closing the valve *A* at the boiler, as shown in Fig. 13, then if the factory valve *B* is opened quickly the steam and water will be blown out of the pipes and there will be no danger of the pipes freezing if the valve *A* is tight.

Another bad practice lies in screwing a valve on to a pipe by means of a large pipe wrench, thereby running the risk of disfiguring the valve; and if the wrench is placed on the end of the valve farthest from the thread in the pipe, the valve seat will be sprung and the valve will leak.

When screwing a valve on to a pipe close the valve and use a monkey wrench instead of a pipe wrench, or if there is an open wrench which will fit the valve use that, as it will not disfigure the hexagon part of the valve.

Fig. 14 shows a boiler feed pump and connecting steam pipe. One day it became necessary to renew the gasket in the flange, and when the bolts were removed the top part of the flange sprang away from the lower part, as shown at *A*. How this flange was got together in the first place, I do not understand, and it was a wonder that it did not break when the bolts were tightened. The pipe *B* was lengthened so that the vertical pipe could be shifted to the perpendicular, which brought the upper half of the flange to the true position.

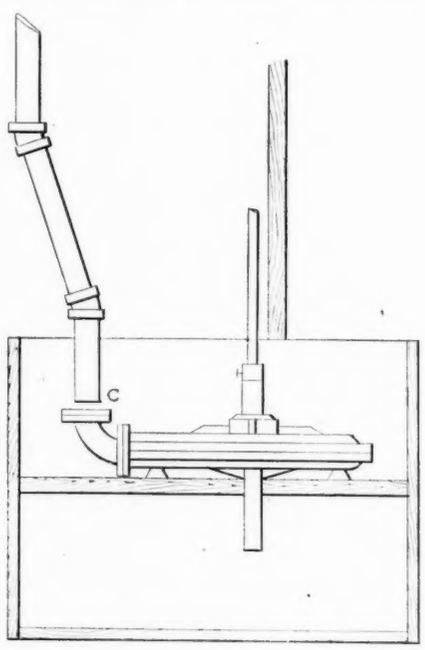


FIG. 15

Fig. 15 shows the relative positions of a vertical belt-driven centrifugal pump and a discharge pipe. The pipe started to leak at the joint *C*, and inside of an hour broke off at the thread and sprung into the position shown. This caused a shutdown of several hours. The break was due to the flange parts not meeting squarely, one of the pipes being too short. I have noticed many cases where the pipe was too short or too long, and to avoid having to cut another piece of pipe, the flange or union was forced to place. Such a joint may not make trouble at first, but trouble is sure to come. Either the union or flange will have to be frequently repacked or the pipe will break.

Another injurious method is to neglect to remove the bonnets from new valves before steam is turned into the pipe, for later when it becomes necessary to replace a disk or regrind it, it is impossible to loosen the bonnet without rounding the corners and disfiguring it. The best plan is to screw a piece of pipe into each end of the valve before putting a new valve in a line of pipe, then tighten the bonnet in a vise; next, give the pipes a few light blows with the hand, and this will loosen the bonnet without damaging it; then cover the threads with graphite mixed with oil, and the bonnet will come off more easily the next time. In valves on which the bonnets are bolted, the bonnet should always be removed before steam is turned into the pipe; then cover both sides of the gasket with graphite mixed

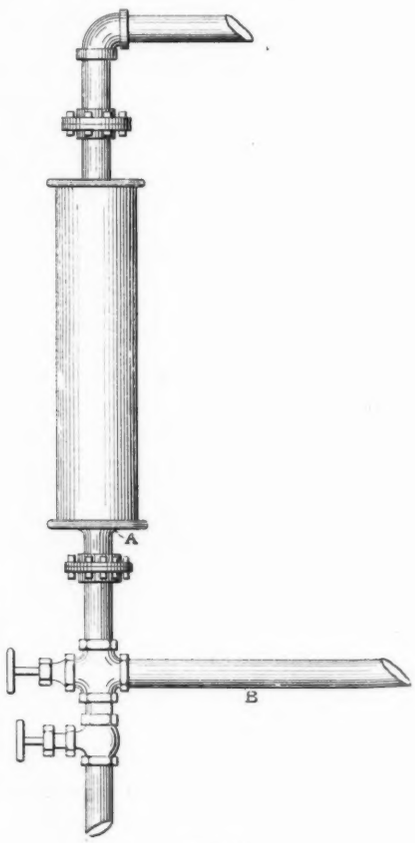


FIG. 16

with oil. When packing a flange joint the gasket should be covered with graphite on both sides; it is also good practice to cover the bolt threads with graphite.

It became necessary to remove the bonnets of a large stop-valve on top of my boiler and when the nuts were taken off the studs it was found that the bonnet

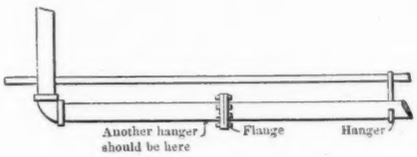


FIG. 17

was so tight that force had to be used to loosen it. In doing this the thread on one of the 1½-inch studs was stripped, making it necessary to take the stud out. In unscrewing the stud the threads in the hole of the valve body became stripped, also. As it was impossible to tap this hole again for a ½-inch stud, we decided to drill and tap it for a ⅝-inch stud,

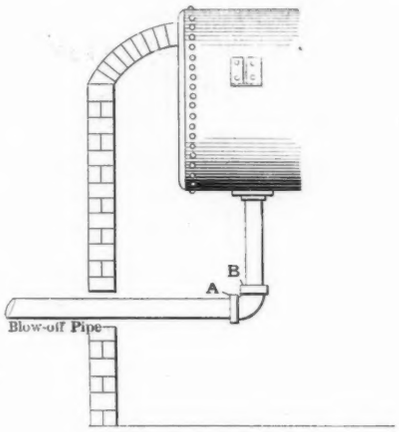


FIG. 18

which also made it necessary to drill the hole on the valve bonnet larger. If the gasket between the bonnet and valve body had been covered with graphite when the valve was put in, all this trouble would have been avoided.

Another bad case is illustrated in Fig. 16. The union on the lower pipe started to leak and when I tightened the union

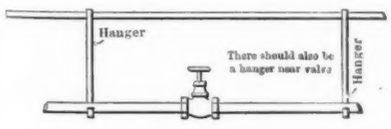


FIG. 19

the pipe sprung a-leak at the joint *A*. When the union was disconnected it was found that the pipe *B* was too short, so that in screwing up the union something had to give way. This was a case where the water column should have been extended to meet the pipe, instead of the pipe meeting the water column. To repair it, the threads in the water column were broken off, the hole drilled and taped for the next larger size, then bushed to the right size.

On a good many water columns unions are used with a gasket between the parts, and this gasket has to be renewed frequently. I have found that the unions of the ground-joint type are best for this purpose, for when once put in there is no future trouble about renewing a gasket. I have often wondered why unions of the ground-joint type are not used in many other places; they do not cost much more and they save trouble, time and worry.

Engineers and steam fitters often complain that pipe joints leak, yet they cannot understand why. Frequently a leak in a pipe joint is due to the pipe not being supported near the joint as it should be. Such a case is shown in Fig. 17 where, when possible, a hanger should be placed near a flange, coupling or union.

It sometimes happens that the blow-off pipe on a new boiler which has been in service for a few months only, starts to leak at the joints *A* or *B*, Fig. 18. This is due to the brick wall settling and pressing the blow-off pipe downward. When taking charge of a new boiler the greatest care should be taken to see that the blow-off pipe is free at all points in the hole where it passes through the wall, as a new wall will settle more or less. The best plan is to have the hole made large enough to allow for the settling of the wall, and then fill the space around the pipe with some material which will "give" when the wall settles.

A few years ago, shortly after two new return-tubular boilers had been put into service, it was noticed that the water in one boiler did not rise in the gage glass when the pump speed was increased. At first we thought the pump was at fault, but at night when the machinery was shut down, and while looking at the fires, I heard the blow-off pipe leaking. The leak was not noticed while the plant was running on account of the noise in the engine and boiler room. It was found that the brick wall through which the blow-off passed had settled on the pipe.

Where there is a valve in a long line of pipe a hanger should be placed near the valve, although I have seen hangers placed a long way from a valve, as in Fig. 19. A valve never should be allowed to carry the weight of a pipe, as it will spring the seat of the valve and cause it to leak.

A comparison of the consumption of coal and gas in the larger cities of the United States shows New York to lead with an annual consumption of 12,500,000 tons of coal, followed by Chicago with 8,245,000 tons. As to gas, New York (Manhattan and Bronx only) consumes 21,000,000,000 cubic feet, and Chicago 12,000,000,000. Of the total coal burned in six cities, in 1906, aggregating 36,428,000 tons, 23,588,000 was bituminous.

Strength of Bolts

By N. A. Carle

Bolts are usually made of wrought iron, rivet and bolt steel, and steel of the following tensile strengths:

Wrought iron, 48,000 pounds per square inch.

Rivet and bolt steel, 56,000 pounds per square inch.

Steel, 64,000 pounds per square inch.

Sizes of bolts are specified as the diameter of the stock from which they are made. The threads reduce the holding power of the bolts to the effective cross-section at the bottom of the thread. This area multiplied by the tensile strength of the material of the bolt gives the total breaking stress of the bolt. This value divided by the proper factor of safety for the service for which the bolt is intended gives the safe working load for which the bolt should be used.

The charts on pages 300 and 301 are designed to read directly the safe working load with various factors of safety for different sizes of bolts with United States standard threads made of wrought iron, rivet and bolt steel, and steel. Chart 1 covers the usual sizes of machine bolts and chart 2 covers anchor bolts.

The data on the number of threads per inch for different sizes of bolts are added for general information, but do not enter into the calculation of the safe working load.

Example

What size of steel bolt should be used to sustain a pull of 6000 pounds with a factor of safety of 8?

Using chart 1, start with 6000 pounds "Safe Working Load" and read up to 8 "Factor of Safety," then across to the line marked "Steel" and down to 1 5/32 inches "Diameter of Bolt." This is not a standard size and the next larger stock size, 1¼ inches, should be selected if the 6000-pound load and a factor of safety of 8 must be maintained. Checking back, the actual holding power of the larger size bolt is found to be 7100 pounds. The next smaller size, 1⅛ inches, would have a holding power of 5600 pounds with a factor of safety of 8.

Example

What would be the safe working load with a factor of safety of 5 of a 4-inch wrought-iron anchor bolt?

Using chart 2 start with 4 inches "Diameter of Bolt" and read up to the line marked "Wrought Iron," then across to 5 "Factor of Safety" and down to 96,000 pounds "Safe Working Load."

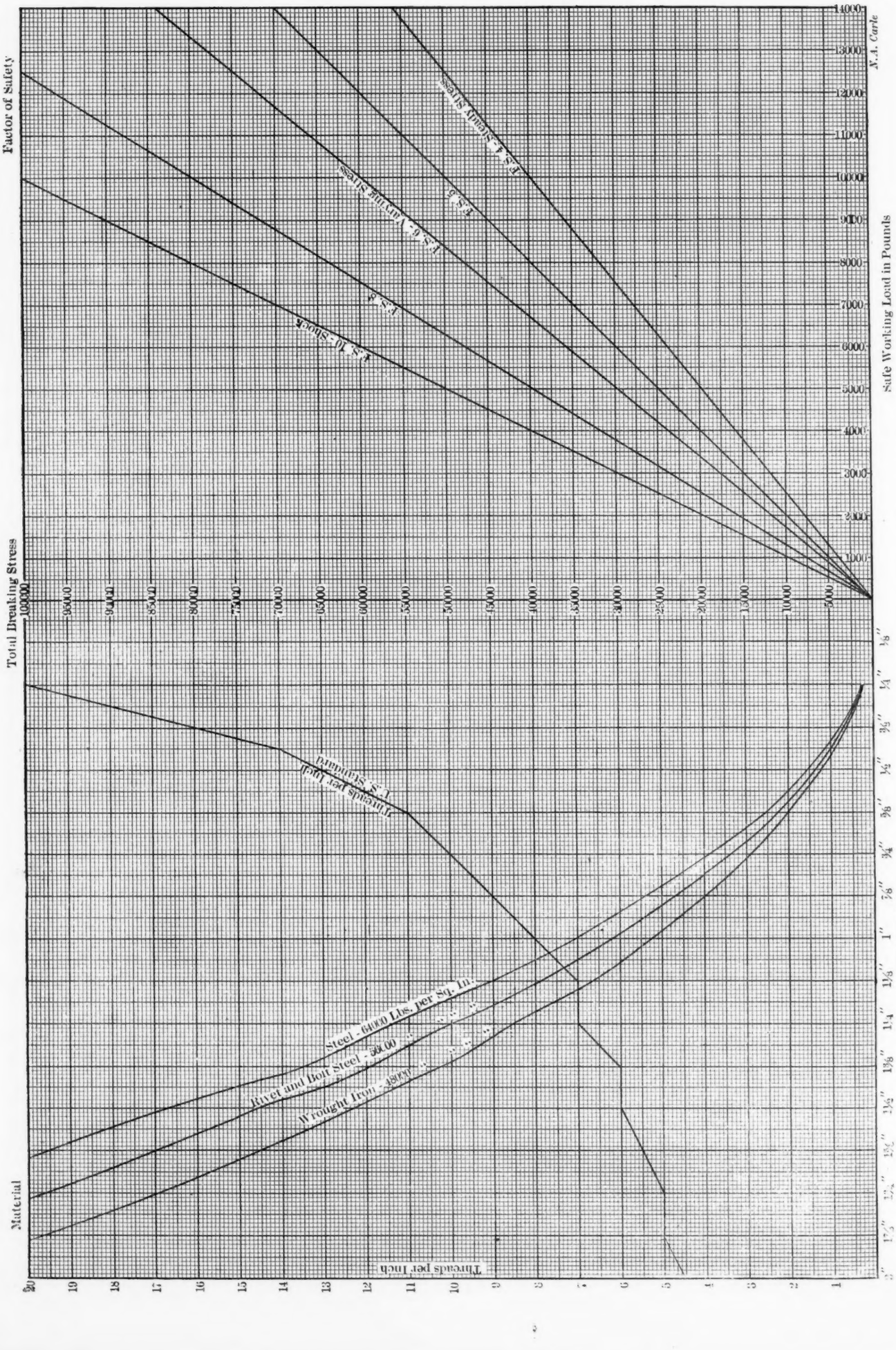

Factor of Safety
Total Breaking Stress
Safe Working Load in Pounds
Material
Threads per Inch
Diameter of Bolt
F.S. 4 - Steady Stress
F.S. 5
F.S. 6 - Varying Stress
F.S. 8
F.S. 10 - Shock
Threads per Inch - U.S. Standard
Steel - 60000 Lbs. per Sq. In.
Rivet and Bolt Steel - 50000 " " " "
Wrought Iron - 48000 " " " "
N. A. Carle

CHART I. FOR CALCULATING THE SAFE WORKING LOADS FOR MACHINE BOLTS

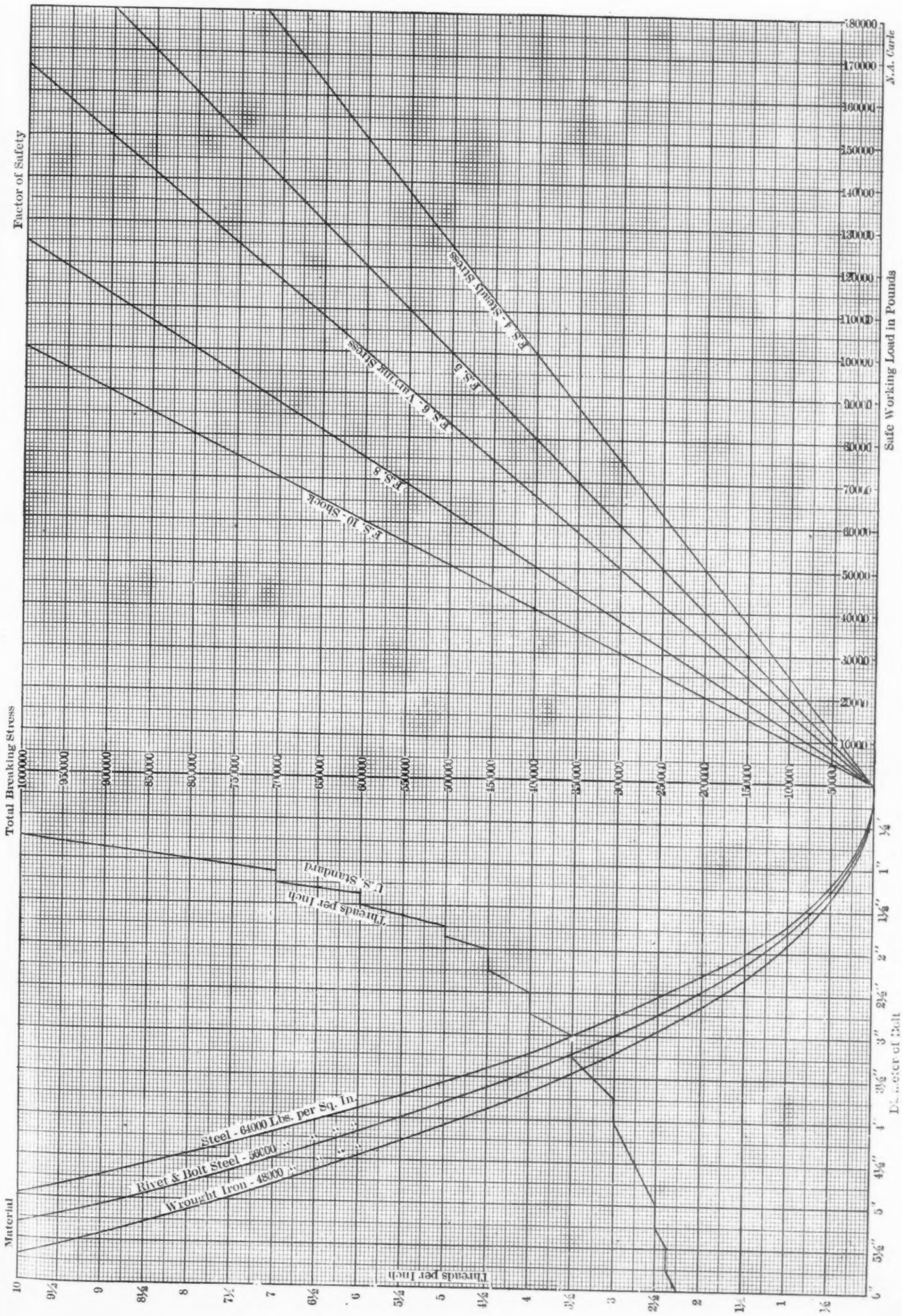


CHART 2. FOR CALCULATING THE SAFE WORKING LOADS FOR ANCHOR BOLTS

How Some Practical Difficulties with Steam Turbines are Overcome

BY EDWARD RUSSELL

In almost every issue of our mechanical and electrical papers we are met with the "wonderful performance" of someone's latest design of steam turbine which is going to revolutionize the engineering world. Following it are lists of consumptions at various loads, some of which may have been taken during the test and the others most likely before the machine has ever felt steam: anyhow, this new machine will never go wrong nor want repairs.

An engineer looks at the heading and

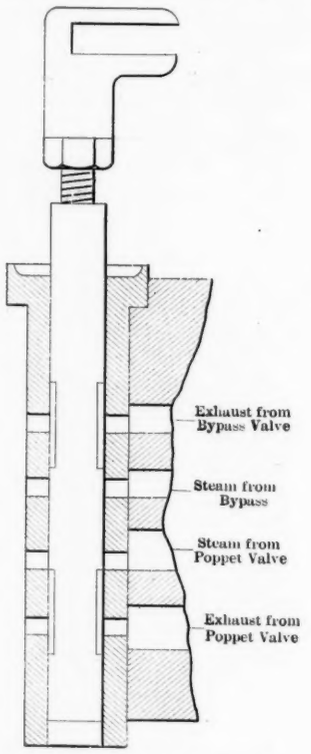


FIG. 1. RELAY PLUNGER

smiles, for, with all our crack salesmen and advertisement preachments, he knows full well there are bound to be troubles and these are what interest him, because as a rule he believes in letting well alone, especially in turbines. In this connection the following will serve to show that even a turbine is capable of giving the engineer in charge quite a lively time:

The first case which we will consider happened in a large English railway traction station containing four 3500-kilowatt Parsons-Westinghouse double-flow machines. This station under ordinary working conditions obtains the whole of its water supply from two artesian wells about 500 feet deep, the water being raised by compressed air. The water originally went through a softener and then mixed with the water in the tank under the cooling towers whence it flowed into the hot-well and was pumped up for boiler feed.

The trouble started one morning after the first turbine had been running a few weeks. At the time the machine was pulling one or two trains which were out for experimental purposes. First, the speed would drop a little and then shoot up in a way which threatened to bring the safety governor out. This would be kept up a few minutes, until it was suddenly discovered that the relay plunger was stuck, and this in the open position which cut all steam off from the machine; consequently, it was falling off in speed. Before anyone could get to it, the speed had dropped so much that the pull on the governor spring had overcome the stuck plunger and the next instant the machine was roaring up to speed again, and it no doubt would have taken the safety governor out this time had not the contractor's man, who had seen such pranks before, cut off the steam by forcing the plunger up to the exhaust position, where it stuck hard and fast.

Once more the tachometer finger commenced to creep from 1000 to 900 revolutions, but this time the figures were not allowed to fall very much, for the small auxiliary valve which controls the flow of steam to the poppet valve was promptly shut, the stop valve throttled down, and the lever which is occasionally used for starting up was brought into commission. This lever consisted of a forging about three feet long, with a piece milled out of one end to allow it to fit around the poppet-valve spindle, and a few inches back another small piece milled out of the bottom side to fit over a small boss cast on the poppet cover to act as a fulcrum.

With this lever the valve was opened just sufficiently to bring the machine up to speed again, and in this manner it was run for a considerable period, while the relay plunger was uncoupled from the rocking shaft, well soaked with paraffin and cylinder-oil mixed, and tapped out with a block of hard wood. On examining it, it had not been galled in the least; what had happened was that the boilers had been priming and a deposit of caustic soda, etc., had been flushed over onto the relay plunger, where it was at once baked hard, on the priming ceasing, by the superheated steam, which had a temperature of about 500 degrees Fahrenheit. The liner was carefully washed out with paraffin and cylinder-oil, the plunger thoroughly cleaned and replaced, and once more put into service. At this time changing over was out of the question, as there was only one turbine in service.

This sticking of plungers went on for a few days, until the plungers had been eased to such an extent that they let steam pass them and hiss into the engine-room, making a deafening noise, with the assistance of the generator's roar. Although there were no stuck plungers, it certainly would not do to leave them like this, so it was decided to make a plunger of fairly good fit and cut a thread on it about 1/16 of an inch pitch, and a full 1/32 of an inch deep, which really meant cutting a spiral groove 1/32 of an inch deep and 1/16 of an inch pitch around the plunger from top to bottom. See Fig. 1. Under ordinary working conditions this answers admirably, the deposit all becoming freed and working off in the groove: or should it show any signs of becoming at all stiff, a little paraffin and cylinder-oil mixed will work down the groove, if put on at the top end of the plunger, and soon put it in perfect order.

Again, when these machines arrived from the shop, the plunger was kept from any twisting movement by small flanges on the little brass block which fitted into the jaws of the relay plunger and gave it its movement. On these being removed the plunger could make about one-tenth of a revolution in either direction, and this movement, of course, greatly aided to keep the plunger free from deposit through the groove.

The machines in question were all

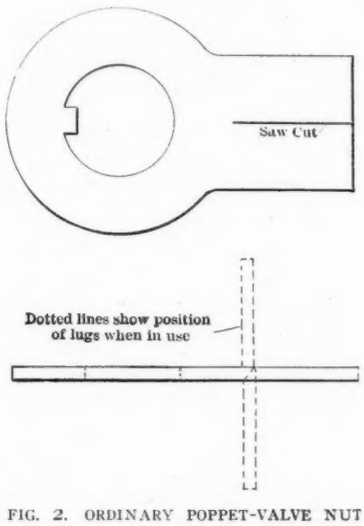


FIG. 2. ORDINARY POPPET-VALVE NUT

fitted with the Brown-Boveri type of poppet valve, long steel bushes and relay plungers, with only one steaming edge, as it were, on the plunger; the bypass or overload valve being worked by hand. In these machines the plungers were very long and a pretty good fit: consequently, one would occasionally gall fast, making it pretty difficult to extract; on more than one occasion the liner being destroyed, and, of course, the plunger.

When there was not a spare plunger the following course was generally successful: The bush would have a reamer of its original bore put through to take off any roughness, or in the absence of a reamer a scraper would be called into service, the plunger in the meantime being oilstoned up, it being inadvisable to use a file on account of minute particles

of the file becoming embedded in the plunger and causing future trouble. When the roughness was taken off both liner and plunger, the plunger would be smeared over with a very thin coating of finely powdered rotten stone and oil, put into its normal working position, and given an up and down movement, accompanied by a slight twisting, for a short time, until a fair polish appeared, when it was withdrawn and thoroughly cleaned, the liner being washed out with paraffin, after which in nearly all cases very little trouble was experienced.

Of course a spiral groove is a great advantage, as the dirt or foreign substance which would cause the galling would get into this groove, and in most cases remain there or gradually work down the groove until it arrived at the port, where the exhaust would carry it away.

Another source of trouble was the slacking back of the nuts on the poppet valve, which would allow it to work up and down on the spindle, with the result that it would often not drop squarely onto the two seats. Of course, this did not matter much on the heavy load, but was a serious matter on the light ones, as the machine would often tend to run up and bring the safety governor out. These were two ordinary nuts of a fairly fine pitch locked together by a washer with lugs, Fig. 2, one of the lugs being bent up, the other down, and the small lug shown on the inner diameter fitted into a groove cut in the spindle.

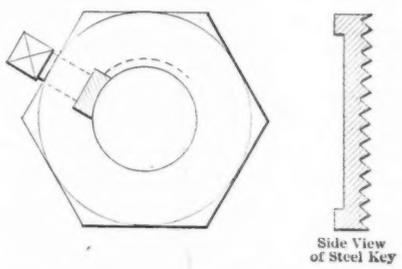


FIG. 3. SPECIAL NUT FOR POPPET VALVE

This was not found to be successful, as the lugs would only stand bending a few times, which meant a new washer if the valve was often opened up. Again, with the constant working of the spindle, the small inner lug would wear down until it allowed a slight movement, which very soon multiplied, with the result of a loose valve. Locking the nuts up tight with a big hammer was then resorted to, which answered fairly well. Another method suggested by the makers was to silver-solder the nuts to the spindle when in position, but this meant a big job—in the first place to solder it up, and then to strip it down again. Eventually a special nut, Fig. 3, was made by the engineer in charge of the station, and up to the present time this nut has given great satisfaction.

Occasionally a poppet-valve spring would break after having been in service for some time, and in nearly every case this was due to faulty material, with a few exceptions due to fatigue, which is not to be wondered at when one considers that the springs are continually compressed and extended; in fact, some making as many as 200 strokes a minute, while the lowest make 108, and in most cases these machines run about eighteen hours a day.

The automatic stop valves on these machines are of the same type as those used by a number of American makers, I believe, and will be understood by referring to Fig. 4. The piston valve V is carried on the rod S which has a piston of two diameters, D^1 and D^2, attached to its top end. Now the steam flows up the pipe P and through the ports P^2, as indicated by the arrows, before passing the poppet valve. When the bottom face of the piston D^2 is on the bottom of the cylinder in which it works, the rod or spindle S is of such a length that the ports P^2 are blinded by the piston valve V, consequently shutting off steam from the turbine. This is worked in the following manner: The bush through which the spindle passes in the flange G is a few thousandths of an inch larger than the diameter of the rod, and it is through this small clearance that the steam has to pass to raise the piston D^2, which, of course, raises the piston valve and opens the ports, putting the valve in its working position.

The steam for shutting the valve, in case of the runaway governor acting, passes up the drilled hole A in the wall of the cylinder into the space O, where the full pressure is always maintained and ready to act on the exposed area of piston D^2. The steam is prevented from passing here by the ground surfaces at M^1 and M^2. A pipe E leads from underneath the piston D^2 to a valve worked by the runaway governor, this pipe having an area many times larger than the opening by which the steam enters around the spindle. Consequently, on the valve being opened the steam will escape much faster than it can enter, so that the pressure will drop in this space while that in the space O is still of the working pressure and acts on the area B with the result that the piston P^2 and the piston valve fall and cover the ports, shutting the machine down.

This piston is generally given about ten-thousandths of an inch side clearance, which is ample in ordinary cases, providing that it does not divert very badly. With this amount of clearance in this case the valves would not work; if they were raised they would not drop again, and if down would generally give considerable trouble about raising on account of the deposit getting between the piston valve and the side of its case and becoming hard, the only way to free it being to pour down a large quantity of paraffin to soften the deposit and keep the valve moving.

Of course, this would have been a pretty serious thing had it stuck and the machine run away, for the safety governor would have been useless. Numerous easings were given but with little effect until they had about fifteen-thousandths of an inch clearance on each side. Even with this clearance they would stick sometimes, so the plan suggested was to

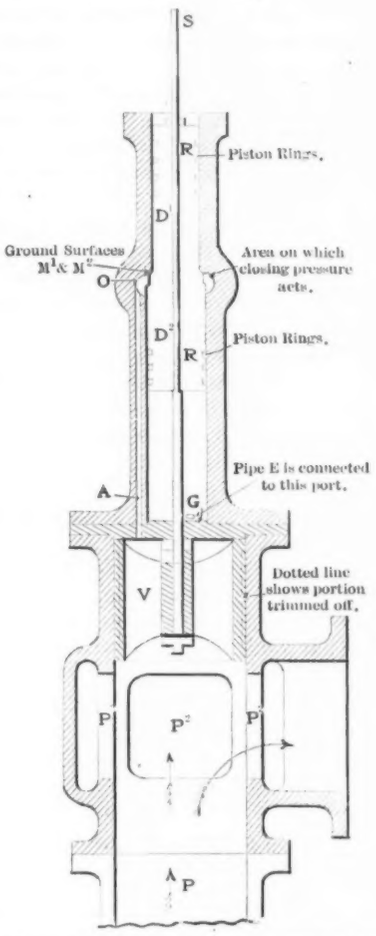


FIG. 4. SHOWING AUTOMATIC STOP-VALVES

put the valve in the lathe and take a very light cut, say 1/32 of an inch each side, off the center of the valve and extend it to within three-quarters of an inch of each end, thereby reducing the bearing surface of the valve without affecting its usefulness in the least. This idea, I understand, has been carried out and is giving every satisfaction. The ideal thing in this case would undoubtedly have been a double-beat valve in a loose chamber, as it would have been impossible for this to become stuck up.

Shaky pipe-lines all remind us,
That the hustling salesman can't
Talk too much about steel fittings
For a modern power plant.

—*Exchange.*

Catechism of Electricity—XXII

ELECTRICAL MEASUREMENT OF CAPACITY

467. *How is capacity measured?*

Capacity is measured by the direct-deflection method previously described, the comparisons in this case being made between the unknown capacity and that of a standard condenser. The standard condenser is of known capacity C, between ⅓ and 1 microfarad, and is charged by means of a battery for a certain time, say 30 seconds, and then discharged through a ballistic galvanometer. Represent by a the first deflection produced. A condenser of unknown capacity C_1 is then charged with the same battery for the same length of time. It is discharged through the same galvanometer and a second deflection a_1 is obtained. The value of the unknown capacity may then be found from the formula $C_1 = C\,\frac{a_1}{a}$.

468. *Show the connections and describe the method of procedure in testing capacity by the direct-deflection method.*

Fig. 103 shows the connections. C will become charged when the lever $h\,r$ of the discharge key is depressed against the lower stop c, for the right-hand pole of the condenser will then receive a negative charge from the battery B by the connection $h\,c\,e\,B$, and the left-hand pole of the condenser will receive a positive charge by the connection $m\,n\,B$. The key being at this charging position, the galvanometer G is set at zero, and when the observer is ready to take a reading, the

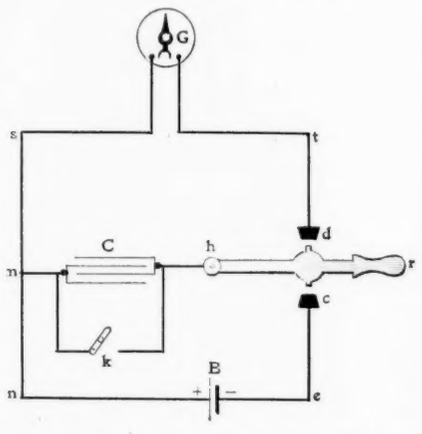


FIG. 103. APPARATUS AND CONNECTIONS FOR THE DIRECT-DEFLECTION METHOD OF MEASURING CAPACITY

discharge should be caused by pressing the lever hr against the upper stop d. When this is done, the poles of the condenser C are placed in metallic communication by the circuit $m\,s\,G\,t\,d\,h$, Fig. 103, and the sudden rush of electricity through the galvanometer causes the kick, or deflection, which has to be noted. While the galvanometer is returning to rest, the condenser should be short-circuited by the key k in order to get rid of any residual charge that may remain. Before again depressing the lever the short-circuit key k should be opened. After the observation of the discharge has been repeated several times, the condenser C is replaced by the unknown capacity and the process repeated.

469. *Suppose a standard condenser of 1 microfarad capacity gave an average deflection of 60, and the average deflection of the unknown capacity was 54, what would be the value of the unknown capacity?*

Substituting the known values of C, a_1 and a in the formula $C_1 = C\,\frac{a_1}{a}$, there results $C_1 = 1 \times \frac{54}{60}$, or 0.9 microfarad as the value of the unknown capacity.

470. *What is a ballistic galvanometer?*

A ballistic galvanometer is an instrument adapted for measuring momentary currents, i.e., currents which last only a very short time. It differs from a D'Arsonval galvanometer in possessing a heavy needle, made nearly spherical. The needle is made heavy in order that its time of vibration will be large, and is made spherical in order that the air resistance will be a minimum. Thus, as the needle swings slowly around, it adds up, as it were, the varying impulses received during the passage of a momentary current, and eliminates to a certain extent the magnetic and frictional resistances which tend to shorten the deflection.

471. *Illustrate and describe a common form of ballistic galvanometer.*

Fig. 104 shows a modern form of ballistic galvanometer. The coils c' and c'', with terminals at m, m', m'' and m''', are supported by means of the polished hard-rubber pillars p' and p'', corrugated in order to give a large leakage surface. The two coils are graded to give the maximum electromagnetic effect for a given length of wire. They must be joined in series, multiple, differentially or singly, by means of flexible insulated-wire connectors. The resistance of the galvanometer can, therefore, be varied according to the measurements which have to be made. The coils are hinged, and can be swung open as in Fig. 104 so that the suspension system e may be easily inspected. By means of the clasps a and b the part d can be securely closed, thus causing the coils c' and c'' to inclose the suspension system. When the part d is closed, a glass window, the frame of which can be seen at r, permits the mirror x on the system to be plainly seen.

The ballistic system e, shown enlarged at the right of Fig. 104, consists of the mirror x and four bell magnets $n\,s$ and and $n'\,s'$, $n''\,s''$ and $n'''\,s'''$, the upper and lower magnets $n\,s$ and $n'''\,s'''$ forming one set, and the central pair $n'\,s'$ and $n''\,s''$ the other. The outer magnets, being the stronger, control the system. The lower magnet has a screw thread cut on it, upon which moves a soft-iron ring w. By screwing this ring toward the poles of the magnet it will make the system more nearly neutral to the earth's magnetic force; this gives a means of varying the sensibility of the galvanometer through a wide range.

472. *Mention a common case where measurements of capacity are necessary.*

All lead-covered cables should be tested for capacity between the conductors and the leaden sheath.

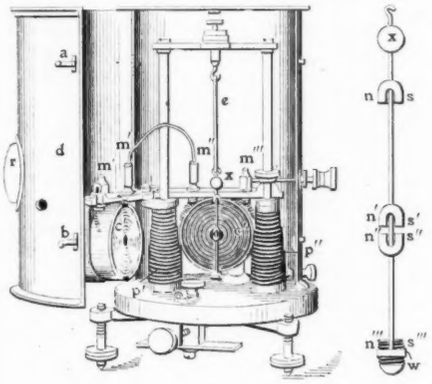


FIG. 104. THE BALLISTIC GALVANOMETER FOR MEASURING TRANSIENT CURRENTS

473. *How are the connections made in determining the capacity of a lead-covered cable?*

The conductors are left open at one end, as at h', Fig. 102, and those at the other end h are bound together and to the sheath. The sheath is connected to h, Fig. 103, and the conductor to be tested is connected to the positive side of the battery B, and also to one terminal of the galvanometer G. The other terminal of the galvanometer is joined to the stop d and the negative side of the battery to the stop c. The deflection produced is that of the unknown capacity, the method of procedure being the same as that described in answer to Question 468.

474. *If the capacity of a certain length of cable be determined by measurement, can its capacity per mile be calculated from this?*

Yes; if l = length in feet of the cable tested; c = its capacity in microfarads as determined by measurement, and C = its capacity in microfarads per mile, then

$$C = \frac{5280\,c}{l}.$$

475. *What is the capacity per mile of a cable whose capacity measures 0.014 microfarad for 750 feet of its length?*

Substituting for c and l their respective values in the formula

$$C = \frac{5280\,c}{l},$$

there results

$$C = \frac{5280 \times 0.014}{750},$$

or 0.099 microfarad for the capacity per mile.

ELECTRICAL MEASUREMENT OF INDUCTANCE

476. *Describe a method of measuring inductance.*

Inductance may be measured with a Wheatstone bridge, condenser, and a variable non-inductive resistance connected as in Fig. 105. In the diagram, *a* and *b* are the ratio arms of the bridge, in each of which is introduced the same amount of resistance. Connected in the x-arm of the bridge is the inductive resistance *L*, the inductance of which is to be measured, in series with a non-inductive resistance *R*, and shunted around these two resistances is a condenser *c*. The resistance *R* is used merely for the purpose of enabling a condenser *c* of practicable size to be employed. The variable-resistance arm of the bridge is denoted by *r*, the battery by *s*, the battery key by *k*, and the galvanometer, which should be of the reflecting type, by *g*.

The test is conducted by varying the resistance in *r*, and if necessary the values of *R* and *c*, until no deflection is given by the galvanometer when the battery circuit is opened. Then, by substituting the values of the capacity of the condenser *c* in microfarads, and of the resistance *R* and *r* in ohms, in the formula

$$L = c\ (R+r)^2,$$

the value of the inductance *L* in henries can be at once obtained.

477. *If the galvanometer in Fig.* 105 *gives no deflection upon opening the key k when there is* 1 *ohm unplugged in each of the ratio arms,* 1 *ohm in the variable-resistance arm,* 1 *ohm of non-inductive resistance in the x-arm, and the condenser*

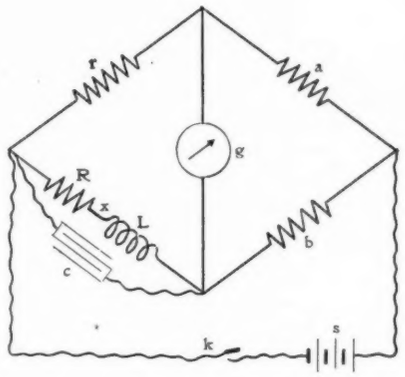


FIG. 105. CONNECTIONS FOR MEASURING INDUCTANCE WITH THE WHEATSTONE BRIDGE

used has a capacity of 0.3 *microfarad, what is the value of the inductance L?*

Substituting 0.3 for *c*, 1 for *R*, and 1 for *r* in the formula $L = c\ (R+r)^2$, there results $L = 0.3\ (1+1)^2$, or 1.2 henries for the value of the inductance.

It is proposed to develop the water power of the Oconee, Flint and Ocmulgee rivers, near Macon, Ga., for the transmission of 50,000 electrical horse-power.

Advantages of the Double-flow Turbine in Traction Work

BY EDWARD RUSSELL

At this writing there are in England two principal and four smaller power stations, equipped with double-flow turbines, with a total output of 105,000 horsepower, and the machines can care for 50 per cent. overload without trouble.

The two principal stations generate power for railway service, which in this instance means not only that they have to deal with very bad loads, but in the lesser station the load conditions at times are about the worst ever heard of, varying between 8000 kilowatts and 1500 kilowatts with two 3500-kilowatt machines in parallel, and with a voltage variation of 2½ per cent. The power station as a rule is lighted from the current generated by the main units, which are three-phase 11,000-volt, through transformers. With the greatest variations of load, flickering of the lamps is unknown when the machines are in parallel, and it is nothing to complain of with only one machine caring for the load.

These machines run at 1000 revolutions per minute and have a tremendous flywheel effect, when it is considered that the drums are 6 feet 6 inches in diameter by 7 feet long and weigh about 18 tons. The steam enters at the center and flows in each direction through nozzles, first striking a row of moving De Laval turbine blades, then passing through a row of guides on to a second row of De Laval blades, after leaving which it starts on the Parsons blades, which continue down to the exhaust end. There are only 40 rows of Parsons blades in the rotor of each machine, or 20 on each side of the center of the drum, preceded by the two rows of De Laval blades.

The first row of Parsons blades is ⅜ inch across, while the widest row is only 5 inches. Of course the first few rows are not very efficient when it is realized that there is a clearance of 0.075 of an inch between the tip of the blade and the cylinder wall, and the same clearance between the guide or cylinder blades and the rotor; so that the percentage of clearance is entirely out of proportion to the blade length, and this of course means a large leakage of steam over the tips of the blades.

Again, the Parsons blades only deal with steam from about 70 pounds pressure per square inch down to 27 inches vacuum, the other 110 pounds per square inch being accounted for in the two rows of the De Laval blades. This type of machine also has an advantage over a single-flow machine in the construction of the rotor, and the short distance between the bearing centers of this part gives the whole a very rigid construction, which is fully appreciated in traction work.

Another great advantage is in the construction of the cylinder, which is built up from three pieces consisting of the center cylinder and two exhaust ends. Around the central section of the center cylinder is a steam belt, having a section of nearly 12x5 inches, which acts as a receiver for the nozzle ring, insuring an equal pressure at all the nozzles. This is also a decided advantage in starting cold, as will be conceived when it is understood that this belt practically covers all the high-pressure stages; so that, on opening the stop-valve the steam must necessarily fill the belt before reaching the nozzle ring, and in so doing tend to warm up, thoroughly and equally, the only part of the machine where damage is likely to result from sudden, unequal expansion. The low-pressure stages and exhaust ends very rarely reach a harmful temperature, unless running non-condensing, the average temperature at the last row of blades under ordinary conditions, having a vacuum between 26½ and 27 inches, being 130 degrees Fahrenheit.

Then, too, in case of a strip at one end of five or six rows there is no need to drop the load nor shut down, as the machines will still develop 1800 to 2000 kilowatts on the De Laval blades. There may be a little more end thrust on the opposite end of the machine, but this would be compensated by the equalizing pipes which connect the Parsons blades on each side of the turbine at their first stage.

Some of these machines, despite all talk, have started from standing cold and have been got up to speed, developing 4000 kilowatts, in less than six minutes. In fact, including the getting of a dry-air pump of the Alberger type under way, and getting the circulating pump into full operation, besides opening an 18-inch valve, six oil valves on the turbine, and a 12-inch valve on the main steam range, the turbine was got up to full speed of 1000 revolutions in less than three minutes.

One morning the last machine installed was to be paralleled with the machine which was carrying the load, and allowed to take one-quarter of the load as a preparatory trying out. When the operator cut the machine in there was the well-known thud, accompanied by pieces of wood flying from the blockings of the armature coils, which indicated that the apparatus was out of phase. There was a load of between 4000 and 4500 kilowatts on the machine in service; consequently, the reverse-current relays on the switchboard acted and cut the new machine out again. The switchboard operator signaled to the engineer to shut down so the machine could be examined, but unfortunately the engineer did not understand what

was required and shut down the machine which was carrying the load, by means of the runaway governor, which could be operated by hand. This naturally caused considerable excitement.

There were shouts from the operating board when the operator saw the voltmeter fall rapidly, with a good chance of a shutdown in sight, but the men who were examining the new machine did not hear the shouts because of the noise of the apparatus. When they became aware what was up, one of the men went over to the old machine and found the speed down to 375 revolutions or thereabout. He at once put the runaway governor in again, opened the stop-valve and in less than three-quarters of a minute the machine was at full speed and voltage. One or two of the sub-stations dropped out of phase on the low speed, but the bulk kept in and the trains were running at full speed again very quickly. The switchboard operator, who formerly worked in large reciprocating stations, stated that he never saw a voltmeter run up the scale at such a rate. If they had been reciprocating units, the mistake would have entailed a complete shutdown, and very likely the same thing would have happened with a single-flow turbine with a low peripheral speed.

The machines in question run in parallel splendidly, even with the steam pressure down to 90 pounds, two of them have successfully carried a load of about 5500 kilowatts, the speed being down to 700 revolutions, with one machine on atmosphere and the other on a very low vacuum, say 20 inches. When running under these conditions there was no tendency to "kick out," as it is expressed.

The machines are controllable from the switchboard through a small motor which operates a weight on a lever, in conjunction with the governor, by which means the speed can be raised or lowered about 10 per cent. The turbo-generators are so well controlled by these motors that in tests that have been carried out, between 4000 and 4300 kilowatts were maintained on one machine, while the other machine was caring for variations of 3000 kilowatts. The engineers simply run the machine up to speed and then pass the word to the operating board. The operator adjusts the speed for paralleling and divides the load through the motors, so that there is very little time lost in paralleling, and none of the misunderstandings which often occur in signals.

On January 8 last two engineers in the employ of Willans & Robbinson, of Rugby, England, were killed by the bursting of a water brake. The drawings were furnished by an American engineer, and the failure of the brake was due to the inadvertent use of cast iron for the drums, instead of cast steel, as the drawings called for.

Some Practical Kinks

By William Kavanagh

Every engineer who has to cut and fit his packing should make a substitute piston-rod. This can be made of wood and of different diameters, corresponding to the sizes of rods found in the plant. Suppose there are four different diameters of

FIG. 1. SUBSTITUTE PISTON-ROD

rods, then the substitute rod will have the appearance shown in Fig. 1. A piece of soft pine wood may be employed for the purpose. If a lathe is convenient, place the piece of wood in it and turn it down to correspond with the diameters of the engine or pump piston-rods, as the case may be. If the distance from shoulder to shoulder be made six inches, and there are four different diameters, the piece of wood will be twenty-four inches long.

This tool will be found to be very handy; instead of waiting for the engine to be stopped in order to obtain the length of the piece of packing needed, the size may be taken from the wooden rod and the packing cut at leisure without danger of one's being burned or scalded. Besides, if the wooden rod is accurately made the packing can be fitted to a finer nicety than could be done by wrapping the packing around the actual, hot piston-rod, trying to locate the required length. Those especially who have charge of compound engines, where the cylinders are superimposed on each other, or even where the cylinders are in tandem, will find this substitute rod a time-saver as well as an accurate tool.

At the plant with which the writer is connected we carry two hundred pounds pressure on the boilers and employ reducing valves. Two of the compound engines are supplied with steam at 150 pounds, while 110 pounds pressure is delivered to the Corliss engines. It was found recently that one of the compound engines was receiving full boiler pressure, the reducing valve being inoperative. Investigation disclosed that the rods supporting the springs were corroded, and that all the valve needed was a little oil in the guides through which the rods passed. The reducing valve is shown in Fig. 2. The corroded parts were at A A^1.

There is a common exhaust main for all of the engines, and included in the line is the elbow at M, Fig. 3, which had a bad habit of breaking; it had to be renewed three times. Different ideas were expressed with respect to the cause of the breaks. Some said it was water-hammer, others said the castings were bad and so on. The cure was easily effected by inserting an expansion joint, as shown at P.

Heine boilers are used and each boiler is fitted with a pop safety valve. Through some accident or because of poor management the safety valve on boiler No. 1 used to blow continuously. No amount of tension on the spring would stop the blowing, so it was determined to take the valve off and ascertain the difficulty. A substitute valve was bolted in position so the boiler could be used in the meantime. Inspection of the old valve showed that the seat was badly cut; more than a dozen grooves had been eaten into it by the action of the steam. The disk was also badly grooved.

One of the engine-room assistants was detailed to grind out the grooves, the method employed being as follows: A piece of pine wood about half an inch thick was attached to the back of the valve disk by means of two ¼-inch tap-

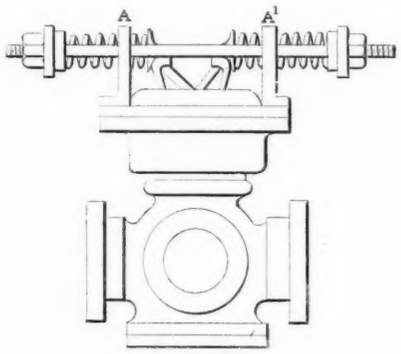


FIG. 2. REDUCING VALVE

bolts, as shown in Fig. 4. On top of this piece of wood was fastened a ½-inch pipe flange into which was screwed a piece of ½-inch pipe of sufficient length to reach about two feet above the bottom of the valve. Diametrically across the valve body was bolted a section of pine board in which had been drilled a hole to coincide with the valve center. The ½-inch pipe extended through this hole, the sides of which acted as a guide for the pipe. A carpenter's brace was fitted to the ½-inch pipe and by turning the brace to either the right- or left-hand, the valve disk was made to assume a corresponding motion.

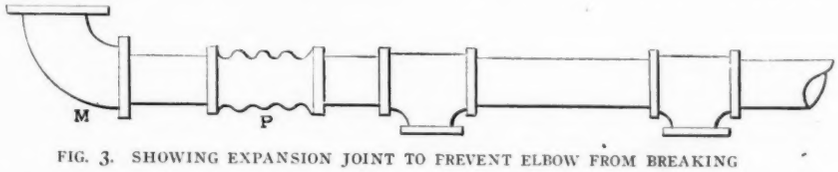


FIG. 3. SHOWING EXPANSION JOINT TO PREVENT ELBOW FROM BREAKING

Ground glass and machine-oil composed the cutting agent. By raising the valve disk the seat could be painted with the ground glass and oil. Then the disk was lowered and its rotation effected by the means described. The grinding process

was kept up until all of the grooves were worn away. The ground glass was then dispensed with and oil was used alone in the finishing process. It took considerable time but the job was so satisfactory that it paid. If a lathe had been at hand some time could have been saved, although in any case the disk would have to be ground to the seat. The oil-finishing process develops a highly polished surface that, in all probability, cannot be surpassed by any other method.

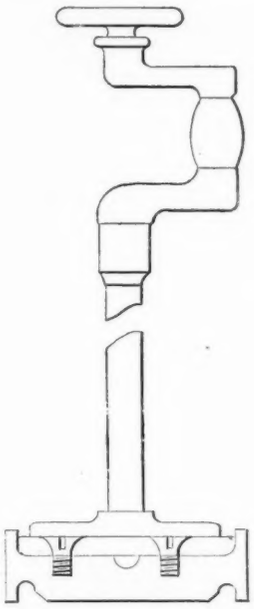


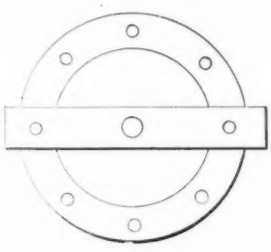
FIG. 4. FOR GRINDING OUT GROOVES

An accident occurred to the stuffing-box of one of our high-speed engines, the stud-bolts being broken off close to the cylinder-head. Owing to the fact that there is a diaphragm in front of the stuffing-box, the stud-bolts could not be drilled out. The stuffing-box is cast in the plug, which is fitted into the cylinder-head, as shown in Fig. 6, and it was withdrawn as follows:

FIG. 7. HOLDING-DOWN BOLT

A heavy piece of timber, for a brace, was bolted across the cylinder-head, having a hole central with the stuffing-box center, as shown in Fig. 5. A ¾-inch iron rod of sufficient length to reach from the stuffing-box to the wooden brace was threaded on both ends and nuts and washers attached. By screwing up the nut at *B*, Fig. 6, the plug was easily drawn into the cylinder. The old stud-bolts were then drilled out and new ones inserted. It will be noticed that the taper or plug looks toward the crank end, so that the steam always exerts a pressure to maintain the plug in position.

The engineers who erected this plant made a serious mistake with one of the Corliss engines. The holding-down bolts were more than six inches too short and to remedy the mistake sockets were screwed on one end of the bolts, into which were screwed short bolts to make up the deficiency. Fig. 7 shows one of the original holding-down bolts with socket and short bolt attached. Later the sockets on the holding-down bolts of the outboard bearing stripped, and it was at first thought that in order to renew those sockets the fly-wheel, shaft and bearing would have to be removed.



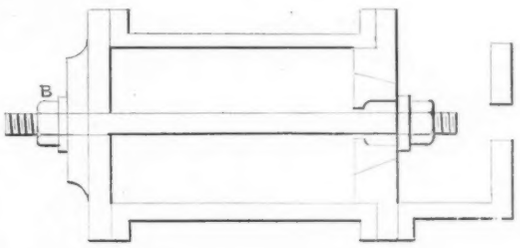


FIG. 5. HOW BROKEN STUD-BOLTS WERE REMOVED FIG. 6

The foundation of the outboard bearing is built in line and close to the buttressed portion of the engine-room wall, as shown in Fig. 8. At *A* is a plan view of the bearing, shaft and fly-wheel. The fly-wheel is fourteen feet in diameter and, of course, the removal of this machinery would entail considerable expense, not to mention loss of time. A section of heavy plank *B* was secured to the wall by means of expansion bolts. A piece of planking three inches thick and twelve inches wide was adjusted from the wall to the bearing *C*. This plank was spiked to *B*, and was sufficiently long to prevent movement of the bearing in that direction.

This part of the problem was easy, but to prevent movement of the bearing in the opposite direction was where the difficulty lay; many ideas were tried out before the plan adopted was evolved. Three 2-inch nipples were cut to fit, as shown at *D E F*, the long nipple *E* being right-and-left. The elbow at *F* was made left because the pipe tongs could be employed to screw up the nipple *E* in the direction of the bearing. A heavy strain was brought to bear upon the nipples *D* and *F* by screwing up the nipple *E*. After this was done the nipple *E* was wedged in place by means of blocks of wood in order to prevent vibration when the engine was running. The engine is now running satisfactorily and to all appearance this bearing will need no further strengthening.

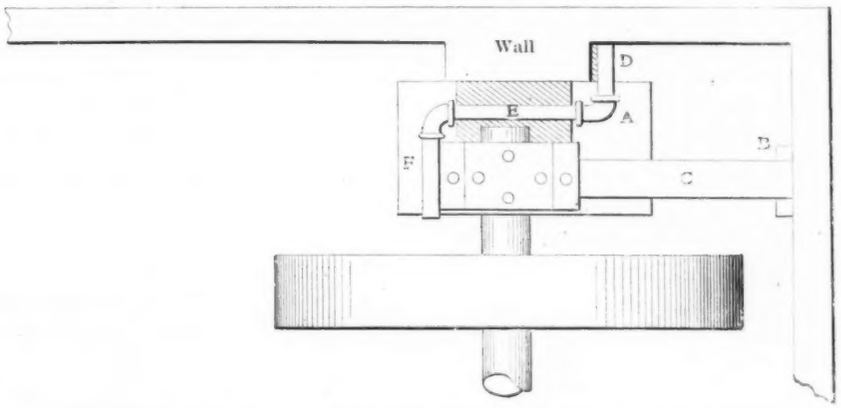


FIG. 8. SHOWING HOW THE SHAFT BEARING WAS STEADIED

Lap-seam Boilers

Prof. Edward F. Miller, of the Massachusetts Institute of Technology, took a decided stand against the lap-seam boiler in a recent hearing before a legislative committee at Boston. "The lap-seam boilers all over the State are certain to explode sooner or later," said the professor. "They are the one type of boiler which gives no warning when about to fail. The other boilers require at least a certain amount of carelessness on the part of the fireman before explosion. Recently," he continued, "I was called as adviser of a company which thinks of taking up boiler-insurance risks, and I advised them not to consider any policies on lap-seam boilers over sixty inches in diameter."

Our correspondents frequently allude to explosions in compressed-air tanks and the pipes leading thereto. It appears that the recent disastrous explosion at Homestead, N. J., of a dynamite magazine holding some four tons of the explosive, was occasioned by the spontaneous ignition of oil in a neighboring tank containing some 15,000 cubic feet of compressed air for use in the tunnel of the Pennsylvania Railroad. Moral: Use graphite.

A Variable-speed Turbine

When steam is discharged upon a turbine wheel with a single row of buckets, as in the De Laval, the speed of the buckets should be approximately one-half that of the jet in order to obtain the maximum efficiency. If the wheel runs slower than this, the steam will be discharged from the exhaust edge of the buckets with more or less residual velocity, and may be reversed in stationary buckets and made to impinge upon additional rows of buckets upon the same shaft, as in the Curtis.

Obviously the greater the number of reversals the slower the efficient speed of the shaft, and a patent has recently been issued in England under which the steam is discharged through a nozzle connected with a set of vanes, which give it one reversal, allowing it to act upon two sets of rotating blades at one speed of the turbine; and through another nozzle, discharging not only through the two rotating and one stationary set of blades as before, but, in addition, through two sets of reversing and two sets of running blades, doubling the number of actions and allowing the speed to be cut in two. With this arrangement the same turbine can be run at practically the same efficiency at two widely different speeds.

Completion of Southern Power Company's Great Falls Station

On March 13 water was turned through the gates, and the water wheels of the Southern Power Company's Great Falls (S. C.) station were operated for the first time. Of the ten turbines six are in service. The machinery equipment of the plant includes eight 3000-kilowatt, three-phase, 22,000-volt Westinghouse generators direct-connected to six horizontal turbines of the Escher-Wyss type and two Hercules turbines. The main dam gives a head of 72 feet and the normal development is 32,000 electrical horse-power, with a maximum of 40,000 electrical horse-power.

Construction was begun in August, 1905. It took eight months to finish the dam at Mountain island and one year to complete the concrete wall at the power house. The record was broken in putting in the concrete on this dam, which is 10 feet wide at the top and 80 feet wide at the base. It is 105 feet high and 650 feet long from hill to hill. There are 100,000 cubic yards of masonry in both of the walls. The power will be transmitted over the only all-tower line in the United States.

Denatured Alcohol

In a paper recently read before the Engineers' Club, of Philadelphia, Henry Leffmann said:

The market price of strong alcohol is from 25 to 40 cents per gallon, exclusive of tax. The figures for its manufacture on a very large scale were taken from the books of a Peoria distillery, averaging 10 years of operation. The average price of corn during those years was 42.36 cents per bushel, and the average yield was 4.76 proof gallons from a bushel. Improvements in distilling increase this yield now to 5 proof gallons per bushel. The manufacturing cost averaged 1.89 cents per proof gallon. For 90 per cent. alcohol, which is the strength used in motors, the cost will be 3.64 cents per wine gallon. Applying the improved methods of distillation, the committee estimates that with corn at the above average price per bushel, and with no material increase in operating, the total cost will be about 18.4 cents per wine gallon. Crude wood alcohol costs 40 cents per gallon to manufacture. If 10 per cent. of this were used in denaturing, the cost of such denatured alcohol would be 20.5 cents per wine gallon. At this price it would compete with gasolene, and in the Northwest, where gasolene is higher and corn cheaper, the competition would seem to be still more active. It must also be borne in mind that the supply of gasolene is limited, while that of alcohol is unlimited.

A very abundant source of alcohol is the waste molasses—especially beet-sugar molasses—the alcohol from which is of inferior value for liquors and medicines, but of equal value with the best alcohol for burning. It is reported by United States officials that in Cuba alcohol is made from this molasses at a cost of 10 cents per gallon. As the by-product of beet sugar goes largely to waste, on account of the inferior quality of the alcohol, the utilization of this will be a powerful factor in the development of the beet-sugar industry.

Mr. Glasser's Tank Indicator Connections

In the March number of POWER, on page 167, was published the description of an electrical indicator for water tanks, by C. H. Glasser. By reason of an engraver's error the diagram accompanying the description showed the ammeters (used as indicators) connected directly across the supply main, which would, of course, short-circuit the main. The correct method of connection is to put the ammeters in series with the lamp bank. The faulty and corrected diagrams are shown herewith.

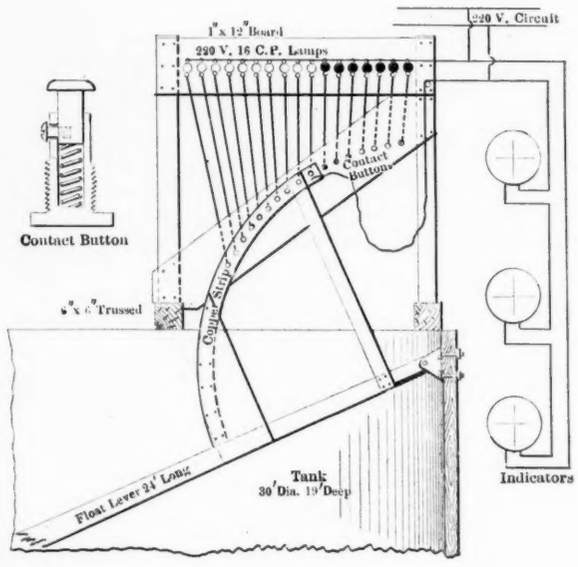
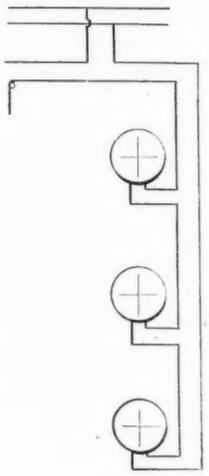


DIAGRAM AS ORIGINALLY PRINTED



CORRECT CONNECTIONS

The United States battleship "Cincinnati" has eight boilers of the water-tube type, which, when tested for efficiency of steam raising, gave the following results: Fires started at 9:40; in five minutes steam formed; in six minutes the gage showed 25 pounds pressure; in nine minutes from starting fires with cold water in the boiler the pressure was 65 pounds, and the pressure rose every thirty seconds 10 pounds; in eleven minutes and a half the pressure was 155 pounds per square inch, ample to get under way and to handle ship as required. According to the *Scientific American*, in twelve minutes and forty seconds there were 215 pounds per square inch on the gage, and the vessel was free to go wherever she listed at full speed.

A Constant Head Viscosimeter

By F. B. Davant

The instrument shown in the accompanying illustrations is the result of a long series of experiments made by the writer for the purpose of producing an apparatus which would give refined and uniform results. This purpose is satisfactorily accomplished, and the instrument is now in use in the lubricant-testing laboratory of the University of Tennessee.

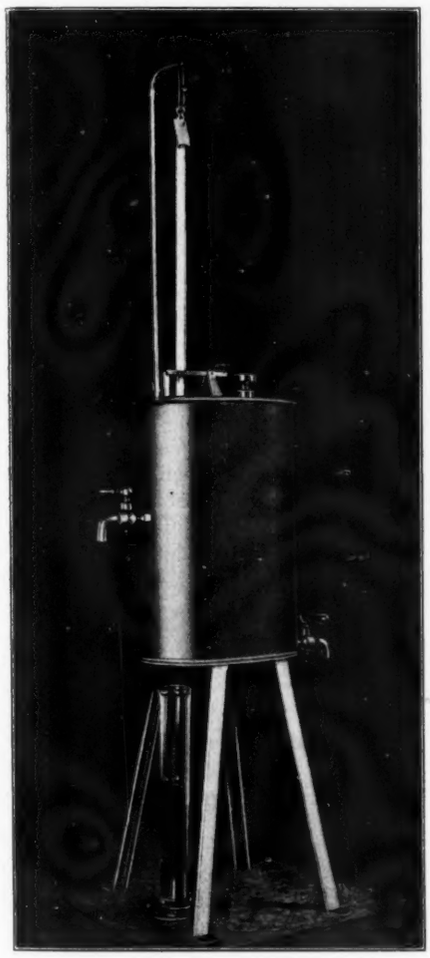


FIG. 1. A CONSTANT-HEAD VISCOSIMETER

The specific claim made for it is that a constant head is maintained above the orifice, and that the oil can be held at any desired temperature throughout the test. Fig. 1 shows the viscosimeter set up for use.

Fig. 2 shows the construction of it in detail, including dimensions. The oil to be tested is contained in *A*, *I*, *B* and *C*, which are entirely surrounded by the jacket forming the outline of the instrument. *C* and *I* are open to the atmosphere at the top, while the reservoir *A* is closed by the cap *d*. When the instrument is in use the oil flows from *A* through the openings *a*, *b* and *c* into *C*, and is finally discharged through the orifice at *g*.

To operate the instrument all the drain cocks are closed, the cap *d* removed, the oil to be tested poured into *A*, and *d* replaced. If the temperature at which the viscosity is desired is above 212 degrees Fahrenheit, oil should be put in the jacket space *W*; if below 212 degrees water may be used. Since *I* and *C* are open at the top, the oil will rise simultaneously with that in *A*, and after *d* is screwed into place *I* and *C* are drained to the dotted line *e—f* (the working head) through *D*.

When the surface of the oil in the air inlet *I* is brought to *e—f*, the oil is held in *A* by the vacuum formed above it. The vacuum is such that the weight of the oil in *A* plus the vacuum (which, of course, is a negative quantity) will just equal the atmospheric pressure. This balance of pressure, which is automatically maintained, prevents a flow of oil into *C* when the orifice *g* is closed, although the level in *C* may be several inches below that in the reservoir. When the valve is lifted by pressing the button *K*, and the oil flows through *g*, a slight fall of level occurs in *C*, and likewise in *I* and *A*. This fall increases the vacuum in *A*, making the pressure in the reservoir less than that of the atmosphere, causing an inflow of the air through the angular notch *h*, sufficient to make the pressure in *A* greater than that of the atmosphere, and forcing a small quantity of oil into *I*, and thence into *C*, from which it ultimately passes through *g*.

It will be observed by reference to Fig. 2, that the interposition of a system of baffle plates forms a very circuitous path for the oil to follow from the reservoir *A* to the orifice *g*. The frequent changes of direction of the oil, thus brought about, reduce its turbulence, bringing it to a state of almost perfect placidity by the time it reaches *C*, while at the same time it is made to alternately rise above and fall below *h* very rapidly, thus checking and admitting the air supply to *A*, and regulating the flow of oil. This process goes on continuously, causing rapid pulsations from the reservoir, but maintaining in *C* a head that varies less than one-sixty-fourth of an inch by actual measurement.

To obtain the most satisfactory results several runs should be made for each temperature desired, and the average found of the readings most nearly corresponding. Every operator should find the water rate for his instrument whether it be stamped on the machine or not, as accuracy of results depends more on the skill of manipulation than anything else.

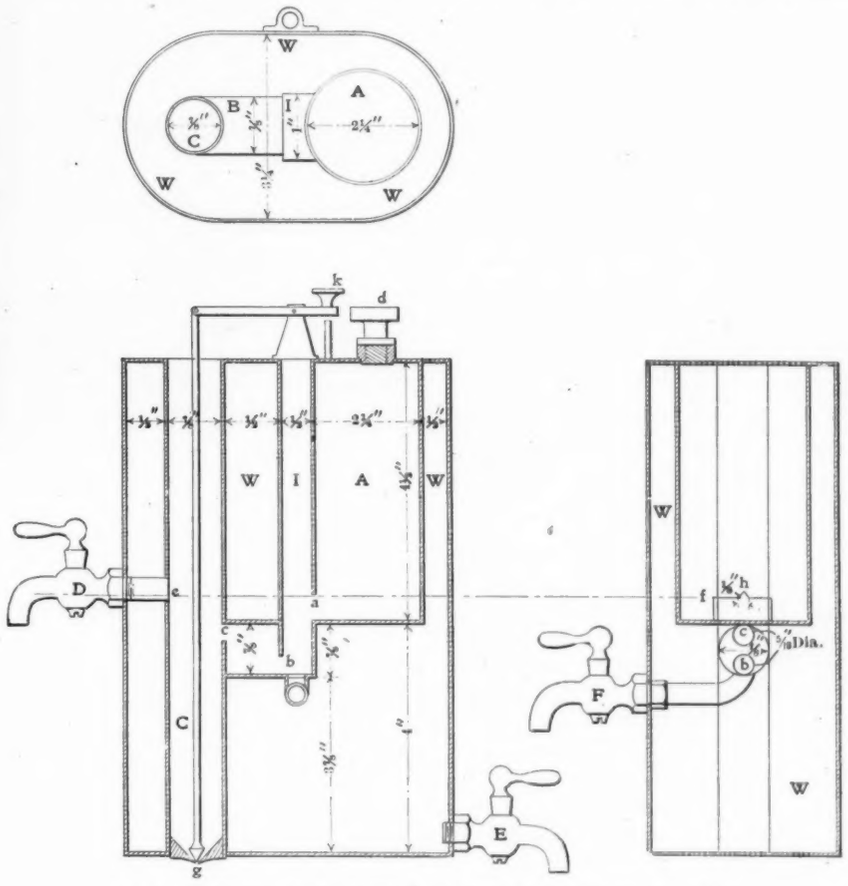


FIG. 2. SHOWING CONSTRUCTION OF THE VISCOSIMETER IN DETAIL

When the test is finished the viscosimeter may be drained by holding *g* open until the tube is free from oil. Since the opening *c* is at the top of *B*, a small quantity of oil is entrapped in *B*, and must be drained off through *F*, Fig. 2. The water jacket is drained through *E*. The temperature of the water jacket can be controlled by a Bunsen burner.

Engineering Societies Building Dedicated

Three of the Four Great National Societies Adopt the Capitol of Professional Engineering Presented by Andrew Carnegie

The middle week of the past month is one which will be marked with red letters in the history of engineering in America. On Tuesday afternoon of that week the building donated to the three great national engineering societies, the American Society of Mechanical Engineers, the American Institute of Electrical Engineers and the American Institute of Mining Engineers, was formally dedicated "to the advancement of the engineering arts and sciences in all their branches."

The main auditorium, providing seating capacity for about a thousand persons, was well filled with members of the three societies when Charles Wallace Hunt, past president of the American Society of Mechanical Engineers, called the meeting to order, using the setting maul used by Mrs. Carnegie in laying the corner stone on May 8, 1906, it having been appropriately adopted for use as a gavel at this as well as all subsequent formal meetings in this hall.

After prayer by the Reverend Edward Everett Hale, telegrams were read from the Presidents of the United States and of Mexico, and from the Governor-General of Canada, the latter of whom was prevented from attending only by a delayed railroad train.

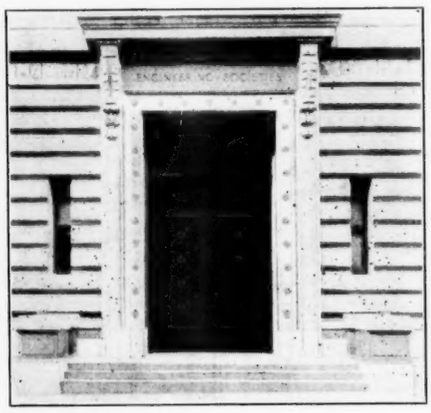
Charles F. Scott, past president of the American Institute of Electrical Engineers, and chairman of the Building Committee, reviewed the events which led to the donation by Mr. Carnegie of one million dollars for the building, the subsequent increase of the sum to a million and a half as the development of the plan called for the greater expenditure, and the transformation of this means into the building which should be the American home of engineering.

Mr. Hunt then placed around the neck of E. E. Olcott, president of the United Engineering Society, representing the founder societies, a tri-colored ribbon to which the key of the building was attached, and formally turned the building over to him in trust for the holding societies. The handle of the key was furnished by Tiffany & Co., with their compliments, and carries a laurel wreath inclosing the badges of the three societies. Underneath a rock crystal lie three grains of the original gold discovered by John Marshall, on Sutter creek in California, in 1848. Mr. Olcott accepted the charge in a few well-chosen words.

Andrew Carnegie, the donor of the building, was the next speaker. He had said, in assuming the cost of the building, that it was not to be a "monument" to engineering. Engineering was not dead enough to need a monument. It was to be a home of engineering filled with a living soul, and he was satisfied with the outcome and full of credit for the architects and builders, for the committee and everybody connected with the enterprise but himself.

The oration was delivered by Arthur T. Hadley, president of Yale University, and dealt with "The Professional Ideals of the Twentieth Century." Its key-note was the incorporation of the ethical with the technical effectiveness.

Among other things he said: "It is not enough that a doctor shall be proficient and successful in the treatment of individual cases. This will make him rich, but the ideal physician deals with disease



ENTRANCE TO ENGINEERING SOCIETIES BUILDING

in a broader way, for its prevention and eradication.

"There are two quite distinct qualities which must be combined in order to secure the best professional service; two quite distinct tests, which work must meet in order to be pronounced first-class. One of these is the technical standard; the other, for want of a better word, may be called the ethical standard.

"This larger view of professional obligations is not so fully recognized as it should be. We have, in the nineteenth century, made so much progress in the technical training of doctors and lawyers and engineers that we sometimes forget that there is need of anything more than technical training. We have let the old idea of public leadership, which was prominent in the minds of the great professional men of past centuries, give place to another and narrower ideal, which is fully satisfied when a man has made himself a technical expert. Many a man of real eminence in his calling deliberately rejects the wider conception of professional duty which I have here indicated. Perhaps he recognizes the claims of public service, perhaps he does not; but in any event he believes that these claims rest upon him as a man, rather than as an engineer or a lawyer.

"There are three professions today which do not regard themselves as servants, but as masters—the financier, the journalist, and the politician. If the engineer and the lawyer accept positions as servants, simply putting their technical knowledge at the disposal of the merchant, journalist, or politician who will pay the highest price for it, it is not simply a confession of inferiority—it is a dereliction of public duty."

On Tuesday evening a reception was held in the auditorium, followed by receptions by the officers of the several societies in their respective quarters.

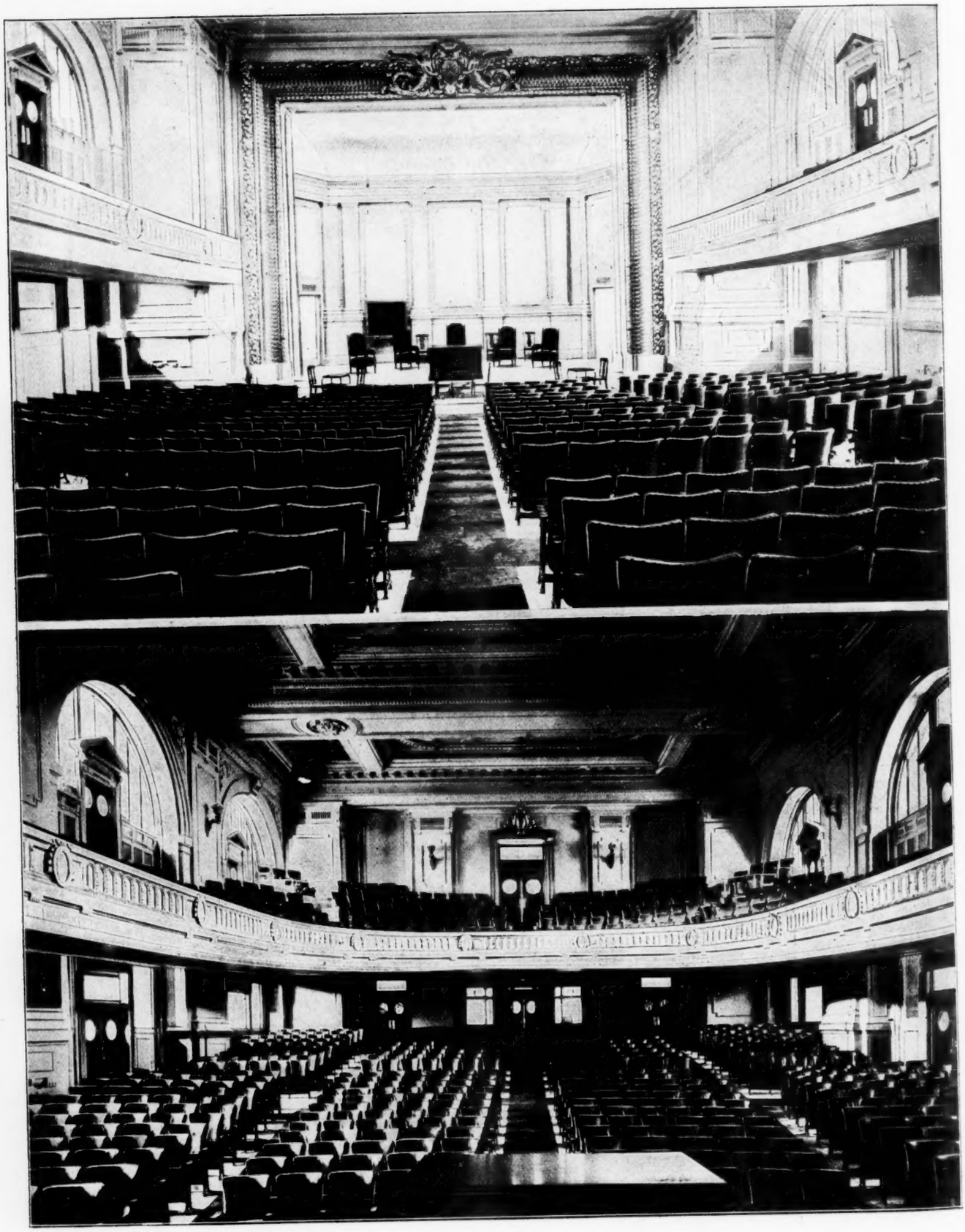
Wednesday, the 17th, was "Founders' Day," being devoted to addresses by the presidents of the founder organizations in which the ownership of the building is vested. The session was opened by introductory remarks from John W. Lieb, Jr., chairman of the Dedication Committee. Dr. Samuel Sheldon, president of the American Institute of Electrical Engineers, then delivered an admirable address in which he pointed out the magnitude to which engineering undertakings had grown. F. R. Hutton, president of the American Society of Mechanical Engineers, was the next speaker. He said that the new building stands as a rebuke to certain economic fallacies, refuting the contention that the labor of arm and brawn is the only real creator of wealth. T. C. Martin, president of the Engineers' Club (the building of which adjoins the main building, and was included by the Carnegie donation), delivered a felicitous poem entitled, "The Little Club Nearby." The final adress was that of John Hays Hammond, president of the American Institute of Mining Engineers.

Greetings from American and foreign scientific societies were read, and the exercises were closed by the presentation of the John Fritz gold medal to Alexander Graham Bell, inventor of the telephone, and of commemorative medals for distinguished services to Ralph W. Pope, secretary of the electrical organization, F. R. Hutton, for many years secretary of the mechanical society, and Rossiter W. Raymond, secretary of the mining society.

ENGINEERING SOCIETIES BUILDING

LOBBY AT SIDE OF AUDITORIUM

MAIN ENTRANCE HALL, ENGINEERING SOCIETIES BUILDING



TWO VIEWS OF THE AUDITORIUM, ENGINEERING SOCIETIES BUILDING

TWO VIEWS OF THE LIBRARY, ENGINEERING SOCIETIES BUILDING

Practical Letters from Practical Men

Don't Bother About the Style, but Write Just What You Think, Know or Want to Know About Your Work, and Help Each Other

WE PAY FOR USEFUL IDEAS

"The Influence of Central-station Loads on Smoke Production"

Dr. C. E. Lucke's article in March POWER on the "Influence of Central-station Loads on Smoke Production" is a valuable contribution to the literature of smoke prevention, but the sub-headline under the title is apt to be misleading. It states a conclusion which is not found in the article itself and can by no means logically be drawn from it. The line reads, "Continuous Smokeless Fuel Combustion Impossible with Rapid Load Fluctuations." The word "impossible" is one which an engineer should be very careful about using. Most of the engineer's work of the present day is the doing of what was considered impossible a few years ago. The line should read, "Continuous Smokeless Combustion with Rapid Fluctuation is Impossible in a Furnace Not Designed to Meet the Difficulties Due to Fluctuation of Load, but is Quite Possible in a Furnace Properly Designed."

Doctor Lucke correctly states the conditions necessary for smokeless combustion, namely, a distilling chamber, a combustion chamber at a high temperature, a supply of air and means for mixing the air with the gases in such a way that practically all the gas comes in contact with the oxygen at a high temperature and remains long enough for combustion, and means for preventing the gas from being chilled by cold air or cool boiler plates before combustion is complete.

Suppose we have a furnace designed for the burning of 2000 pounds of coal per hour. It has a distilling chamber so large and so hot that the coal is not pushed out of it into the coke-burning chamber until all the gas has been distilled; it has a combustion chamber so big and so hot that with a sufficient supply of hot air all the distilled gases are completely burned. Such a furnace is entirely smokeless for all rates of combustion below 2000 pounds. If it is attempted to burn more than 2000 pounds per hour, then the coal will be pushed on to the coke-burning grate before the gases are distilled and the rapid production of smoky gases will take place in the chamber which was designed only for the burning of coke, not for the distillation of coal. These smoky gases may still be burned smokelessly if they are supplied and thoroughly mixed with hot air in the hot combustion chamber, but if either the air supply or the means of heating it or the size of the combustion chamber is insufficient, then the furnace will not be smokeless for any greater rate of driving than that for which it was designed.

If the loads of the electric station are expected to fluctuate, say, between the extremes of 4000 to 2000 horse-power inside of a few minutes, then the electric generator and the engine must be designed so as to carry the maximum load without slowing down; the boiler must be designed to carry the overload without a serious drop of steam pressure. Why should not the furnace in the same case be designed so as to carry the maximum load without making smoke?

The chief mistake made in the building of power plants in cities, as far as the smoke problem is concerned, is the neglect of the designers to provide enough space in which to build the furnace. They have assumed that no extended furnace nor dutch ovens were needed for Roney stokers under Babcock & Wilcox boilers. They are not needed if the stokers are driven slowly and an arch is built over the upper part of the stoker, but for overload conditions either a dutch oven should be built in front of the boiler or else the boiler setting should be so altered as to make sufficient room for the several parts of the furnace, the distilling chamber, the air-mixing chamber and the combustion chamber, each one of them of sufficient size for overload conditions. When a furnace is so built and properly operated with a sufficient supply of air on top of the bed of fuel during the times of maximum rate of coal feeding, then smokeless combustion with fluctuating loads will be no longer impossible.

Syracuse, N. Y. WM. KENT.

Two Important Points in Gas-engine Cooling and Lubrication

In the operation of the 2000-horse-power blowing engines, working at the Lackawanna steel works, Buffalo, on blast-furnace gas, it has been found that instead of taking the cooling water from the lake directly to the cylinder jackets it is much better to take it from the piston-rod discharge through the jackets of the cylinder heads and finally through the cylinder barrel jackets. The temperature of the water as it comes from the lake in winter ranges from 35 to 40 degrees, Fahrenheit, and when this water was pumped into the cylinder jackets direct there were all kinds of trouble from unequal expansion and contraction, cylinder sweating, etc. As soon as the connections were changed so that the water for the cylinder jackets passed first through the pistons all these troubles disappeared. The temperature of the piston cooling water discharge is about 80 degrees in winter, and the cylinder jackets heat it up to about 140 or 150 degrees, Fahrenheit.

Considerable trouble was experienced at first, too, with the lubrication of these engines. The lubricators were of the force-feed type with sight feeds inclosed in short glass tubes. The mechanism itself was entirely efficient, but it was found impossible to get satisfactory lubrication in the cylinders. The lubricator was so arranged that its pump gave an impulse only when the oil outlet in the cylinder wall was covered by the piston, and this was expected to cause the delivery of oil to the piston barrel, as should be done. Notwithstanding this, it was found that oil was being delivered to the cylinder when the oil outlets were not covered by the piston; the oil was merely burned and the residue scored the cylinder wall.

After a great deal of trouble with the engines and worry over the problem of lubricating them, the cause of the inefficient lubrication was found to be air in the delivery pipes between the oil pump and the cylinder. This air evidently leaked in at the glass tubes surrounding the sight feeds, and formed a cushion between the cylinder and the oil pump; consequently, when the engine piston covered the oil holes in the cylinder wall, the oil delivered by the pump was kept back by the cushion of air, and as soon as the engine piston cleared the oil holes, this cushion was relieved and the oil was delivered into the cylinder behind the piston, where it did harm instead of good. Of course, some oil was delivered against the piston barrel in the intervals while the air cushion was forming, but this was insufficient for proper lubrication.

Substituting a plain oil pump without any openings whatever between the pump and the cylinder cured the trouble entirely. THOMAS BROCKLEBANK.

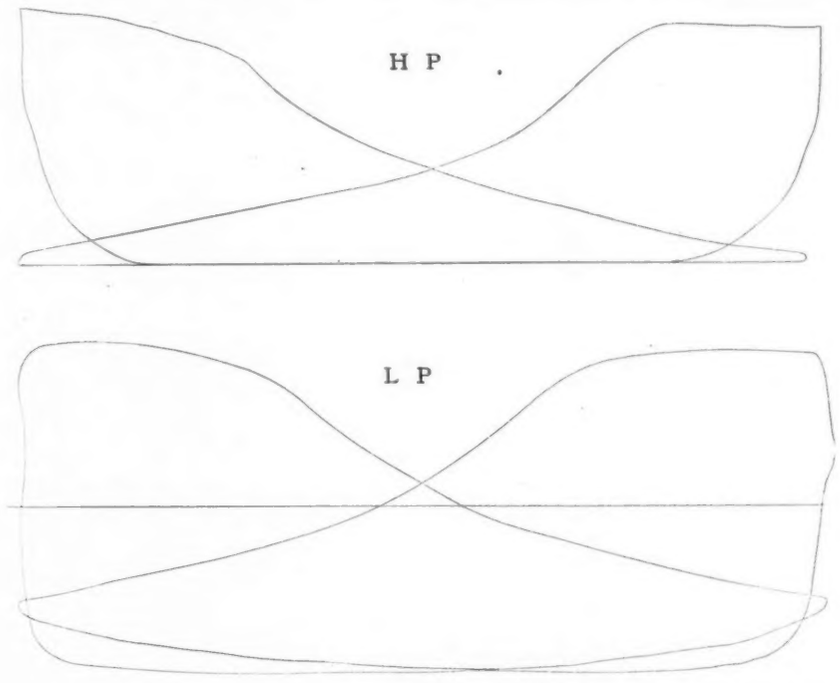
Buffalo, N. Y.

Compression in a Tandem Compound

W. E. Crane, on pages 183 and 184, March POWER, deals with my letter on "The Effect of Compression." His views are largely shared by myself, as he will observe by reading my letter carefully. He says: "The man who gives compression sufficient to check the wheel is using coal to take energy out of the wheel, which has already cost him good money to have put in." Correct. The purpose of my diagram and the explanation accompanying it was to show that when we go too far we are sure to get this effect and its attendant waste of power. Also: "The area of the high-pressure piston is about one-fourth that of the low-pressure piston, and if the high were depended upon to receive all the compression it would necessarily be very high." I said: "The expert who recommends such freakish practice is at variance with all designers and builders of compound engines," etc. He says again: "With the same engine and *the same cut-off* (the italics are mine), when the compression is high the cut-off corner will be rounded and the terminal pressure will be higher, thus losing something in economy, and this could be better afforded in the high-pressure than in the low-pressure." Now under such conditions the cut-off *would not be the same* with abnormally high compression as with a proper amount, because the braking effect of high compression would *put more load* on the engine and the governor would act to cause a *later cut-off*, just the same as if the increased load were due to any other cause; and later cut-off would, of course, result in higher terminal pressure.

As to the sacrifice suggested on the high-pressure cylinder, I do not see that it should be necessary to consider it in ordinary practice, and under ordinary conditions. This would only be necessary where the engine was operated at a higher speed than intended by the designers. The cut-off corner being "rounded" is due to speed, and not to the point of cut-off, since the valve action should be the same. When the piston is moving through its mid-travel we know it moves faster than near the end of the stroke, and it is this greater piston speed that causes the "rounding" of the point of cut-off that Mr. Crane refers to, because the piston moves farther during the time required for the valve to close.

As stated in my first letter, the discussion was prompted by the contention of a gentleman, who was considered "expert," that "more compression should be had in the high-pressure than in the low-pressure cylinder of a tandem compound engine." This expression came from him while considering my criticism of a set of diagrams taken from a 14x28x36 tandem compound-condensing Corliss engine, running at 101 revolutions per minute, and to make the matter more clear, I herewith submit the diagrams.



DIAGRAMS FROM A 14x28x36 TANDEM COMPOUND-CONDENSING ENGINE, 101 R.P.M.

It was contended that "late exhaust closure on the low-pressure cylinder made possible a realization of the benefit of the vacuum until late in the stroke." I pointed out in my criticism that earlier exhaust closure and earlier release would give better results, and that the gain on one end due to late exhaust closure was more than offset by the loss due to the late realization of the benefit of vacuum in the other, due to tardy release, so that the back-pressure line does not fall to a normal position until the stroke is almost completed. I still believe my criticism was correct.

WM. WESTERFIELD.

Charleston, Mo.

"The Steam Plant of the Wood Worsted Mill"

When I read the article in February POWER on the Wood Worsted Mill I thought it an eminently common-sense arrangement and I intended to write my approval. Afterward I concluded to wait and allow some better man to compliment Mr. Diman, but to my surprise I find that all of the good men are inclined to "hand him a lemon."

There may be many boilers with better steaming economy than those of the return-tubular type; but they are not doing business in this "neck of the woods." There are in use in this vicinity several makes of water-tube boilers, horizontal, vertical and cast-iron sectional, and at our plant vertical fire-tube boilers are being operated with fair results, but we believe we can do better with horizontal tubular boilers. I have never seen the equal of the latter, if properly set; much depends upon the setting, of course.

I do not know much about stokers, but I do know that a No. 7 scoop shovel with a good man attached thereto will make a good showing. Also, the man will look after the water, put in a gage glass when needed and shine up the steam gage occasionally. If his boilers are off duty, he could do other work to advantage, too, and if out of order himself he will pay the repair bills. In my experience the man stoker has given the greater satisfaction.

As to coal conveyers, a good horse and cart undoubtedly form a grand combination, but I think a narrow-gage railroad and dumping cars, with a horse for power, are better. The horse on a good rail will haul at least three times as much as the horse in a cart. Besides, if the road is elevated in the boiler room and built just the right distance from the fire door, the coal can always be dumped in the right place and without any shoveling from the cars. The cost is less than for an overhead conveyer and it will outlast any other arrangement of its capacity.

The high-pressure engine-exhaust system is a rather new thing and I would like to hear from others who have tried it. I think it may be all right, however.

Exeter, N. H. A BEGINNER.

I am a firm believer in the water-tube boiler, my preference being based upon a large experience with both the fire-tube

and water-tube types. I learned my trade in a shop where fire-tube boilers were built. Also, as a boiler erector and steam engineer I have had a great deal of experience all over the United States, from which experience I am inclined to believe that about the only real large corporation I know of which still adheres to the fire-tube type of boiler is the American Woolen Company.

On the editorial page of the April number is given a table which I think should be explained, as it is misleading. Too many engineers look upon flue-gas temperatures as indications of the degree of economy attained, and the low temperature given in the table would mislead such. These temperatures are taken at the economizer, which is more than 200 feet away from some of the boilers, and therefore they are no indication whatever of the temperature at the boilers. Again, the high percentage of oxygen would indicate very poor firing, or large air leaks, either of which would tend to lower the flue-gas temperatures. Considering the quality of coal used, I cannot believe that the boilers were showing 65 per cent. efficiency at the time these analyses were made.

Well-designed and properly set boilers should give results in which the flue-gas analysis would show CO_2 averaging as high as the free oxygen shown in this table; and the free oxygen should not be shown to average as high as CO_2 in the table.

In some of the correspondence reference has been made to the ultimate capacity of water-tube and fire-tube boilers, and I would like to suggest that in discussing this subject the actual square feet of heating surface and grate surface should be considered. While Mr. Diman rates his boilers at 200 horse-power each, the rating is based on 15 square feet of heating surface per horse-power; with proper grate area I would consider it possible to get almost as much horse-power out of such a boiler as out of a water-tube boiler having 2000 square feet of total heating surface and rated on the basis of 10 square feet of heating surface per horse-power.

C. H. JOHNSON.

Allston, Mass.

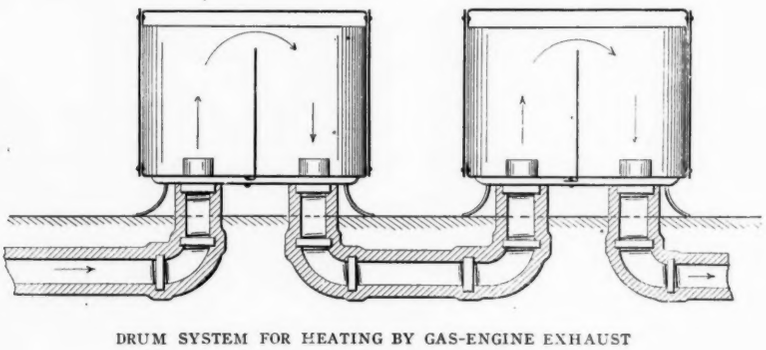
Heating by Gas-engine Exhaust

In reply to R. Manly Orr's request in March POWER, for suggestions how to use the gas-engine exhaust for heating the blacksmith shop, I think he could accomplish it, if the shop is not too high, by leading the exhaust pipe into a series of large iron drums, each divided into two sections by a central partition rising about three-quarters of the hight. The gases would flow up one side of the partition and down the other throughout the series, as indicated by the accompanying sketch. The exhaust pipe could be well insulated and placed underground.

WILLIAM CHADDICK.

Chicago, Ill.

There is absolutely no question but that the gas engine in moderate powers, or even in large powers, is more economical than the steam engine, but it has been found very difficult to make any gain in economy by installing a gas engine, since it is necessary also to use steam for the heating. In Mr. Orr's case, however, stated on page 180 of the March issue, I believe that, as his factory is small, he can utilize the heat from his exhaust to very good advantage. The best way to do this is to install a large muffler, inside of which should be placed a nest of pipes, through which the water of the heating system should pass. These pipes would take the place of the ordinary generator in a heating system where coal is used. The rest of the system can be piped as in a hot-water system, with an expansion tank for the hot water to rise to, whence it would run through the radiators back to the heater, making a complete closed circuit.



DRUM SYSTEM FOR HEATING BY GAS-ENGINE EXHAUST

The sizes of the pipes, radiators and expansion tank depend solely upon the amount of heating surface required, which, in turn, is determined by the size of the room, exposure, glass surface, and other conditions which tend to keep the room cool. One reason why the application of gas engines in general has not been more successful is due to the problem of heating the buildings.

HENRY D. JACKSON.

Boston, Mass.

"Wants to Know About Motor Drive"

On page 259 of April POWER H. G. S. asks for information regarding motor drives in his machine shop. He speaks of a 3-horse-power motor on a lathe, a 2½-horse-power motor on a drill press and a 5 horse-power motor on a pipe machine, all to be operated on a 500-volt direct-current circuit. There is no reason why these motors should not be employed. The motor on the lathe can be mounted directly over the headstock, using a chain drive of the Morse type, preferably; that on the drill press can be mounted directly on the frame of the drill, with a spur gear driving a planetary gear on the drill spindle, or by chain drive direct to the drill, the chain taking the place of the belts. The 5-horse-power motor on the pipe machine can be attached directly to the frame of the machine with a chain drive, as in the case of the lathe.

Chain drives are much more efficient than gears, are quieter and will give excellent satisfaction. All of these motors should be of the shunt-wound type designed for moderate speed.

If H. G. S. is looking for a variable-speed motor it would be advisable for him to consider the installation of a double-voltage machine. If, however, he wishes to vary his speed through the ordinary cones and belts, a plain shunt-wound motor of constant speed will give the best service. This matter, however, is one which it is far better to take up with the manufacturer of the machines, or else put it into the hands of a competent consulting engineer, who will arrange the motors to give far better results than H. G. S. can obtain himself.

HENRY D. JACKSON.

Boston, Mass.

Answering the inquiry of H. G. S., in April POWER, page 259, regarding motor drive for machine tools, I wish to state that I am unqualifiedly in favor of individual motor drive. My experience with this form of power for several years, during which time I installed a large number of individual motors—seventy of them having been placed in one shop—has been so universally satisfactory that were I to make an installation of my own, it should comprise individual motor drive throughout, as far as financial considerations would permit.

It is, of course, somewhat of an open question as to whether it pays to adopt motor drive, particularly direct-connected drive, for old types of machine tools, since the design of many of these tools is such as to require new patterns to adapt them to direct motor drive, which makes it rather expensive.

Another difficulty likely to be encountered is that many of the older types of

tools are not sufficiently heavy to withstand the strains of positive motor operation.

Another and more serious factor is the item of first cost; and this I have found to be the great deterrent when broaching the subject of motor drive to the man who pays the bill.

Looking at it from every point of view, however, I can see no reason why H. G. S. could not use motor drive on all of the machine tools he mentions. With the exception of the pipe machine, they are the same class of tools upon which the writer has used direct drive with manifest advantage and saving of expense.

As regards variable speed and reverse operation on 500 volts, this is comparatively an easy matter. There are reversible motors on the market which are susceptibe of shunt-field speed regulation, speed variations of from 50 to 80 per cent. being common. Personally, I prefer the street-car type of controller to the resistance-box type. The former is now obtainable with the reverse feature, which makes it a very handy and efficient device.

A. P. H. SAUL.

Buffalo, N. Y.

"Should a Corliss Engine Run Under?"

I have had charge of engines which ran in both directions and my experience teaches me that the friction is greater when running over, for the reason that the weight of the crosshead, half of the connecting-rod and half of the piston-rod is acting as a counterbalance to the thrust due to the angularity of the connecting-rod when running under, while this weight is added to the thrust when running over. This weight will be found to be considerable on a 300-horse-power Corliss engine and it will increase with the size of the engine. If the guides are not parallel they should be made so, for the direction in which the engine runs is not determined by the condition of the guides. As to the adjustment of the crosshead shoes, it is better to have them so that they cannot "slap" or "knock," without regard to the direction in which they run.

All will agree upon the importance of keeping the piston-rod as nearly in alinement with the cylinder as possible at all times, and I have found that the crosshead shoes require more frequent adjustment when running over than when running under. The fact that they will run over without slapping when passing centers is no argument in favor of running in that direction, but rather against it, for the crosshead is more likely to get "down," before the engineer discovers it, by the misbehavior of the piston-rod packing.

E. N. G.

Alton, Ill.

"The Barometric Condenser"

In connection with the interesting article on barometric condensers in the January number of POWER, I would like to call attention to an old device patented in Germany, illustrated and described in a specification dated October 20, 1888, a little more than eighteen years ago. This German patent (No. 50,062) was granted to Julius Schwager and embodies the air-siphoning device shown in the "Tomlinson" condenser, described in the article referred to. The idea of taking air from under a spraying umbrella will also be found in later constructions. The old Schwager condenser has the advantage of being a "counter-current" device, therefore has a higher possible efficiency than the direct-current apparatus illustrated in the January number. The old device, perhaps, did not come into more general use because it is found desirable and of commercial advantage to use a dry-air pump in systems where there are several engines and a considerable length of exhaust main, because of air leakage, which is usually in excess of the air-suction capacity of any siphon that may be formed in a barometric tube. In such cases it is very desirable and essential to not only cool but to thoroughly dry the air before it leaves the condensing chamber. The essentials for success in a condenser are analogous to those of a boiler, in which the nature of the work done is reversed.

It is surprising how little serious study has been given by engineers to the details of condenser construction, particularly when considering that nearly every operating engineer is an expert critic on the faulty construction of a boiler. In a boiler he will insist on having perfect circulation, ample steam space, and maximum heating surface consistent with good design. He insists on the greatest possible simplicity and would immediately reject any boiler-feed heater that did not operate "counter-current" wise. However, when selecting a condenser he nearly always asks only for a given vacuum regardless of barometric readings or other conditions. Of course, the ambitious salesman promptly guarantees an ultra refinement bordering on the impossible and regardless of barometric and temperature conditions, always carefully specifying "an air-tight system," well knowing that even an air-tight cheese-box will give a maximum degree of rarefaction, providing the system is absolutely air-tight and a sufficient quantity of water is provided.

It has been found necessary in rolling-mill practice, because of air leakages, not only to siphon as much air as possible down the barometric tube, but in addition to provide an effective means of pumping air out of the condensing chamber. It has been found best not to reduce the downward velocity of the condensing water after once leaving the cataracts or sprays, which naturally cause the water to be thoroughly charged with air that would otherwise again rise into the condensing chamber. The latter must be sufficiently large to readily permit a thorough separation and cooling of the air before it is drawn from the condenser, just as steam is separated from the water and thoroughly "dried" before leaving a properly designed boiler. A large number of condensers literally "slop over" their condensing water into the dry-air pipe from want of proper circulation, precisely as a badly designed boiler carries water over into the steam mains. To avoid wrecking the dry-air pump, because of this defective construction, an auxiliary barometric tube and water-separating device are provided and controlled "automatically" by another device to cause such condensers to return to their normal condition of operation. Such construction would not be tolerated in boilers for a moment by engineers who accept it with good grace in condensers. No doubt engineers will appreciate those timely articles covering the study of condensers, which seemingly has been neglected by many persons purchasing such apparatus.

CHAS. L. HEISLER.

St. Mary's, Ohio.

Improvement in Furnace-door Liners

In the sawdust-burning furnaces in use here the furnace-door liners do not last very long. As usually made the distance pieces which hold the liners out from the door are cast on the plate. The holes for the bolts pass through these pieces, and it the holes do not exactly match the holes in the door, it is necessary to cut out more of the door.

We had a set of these distance pieces cast separate from the liners; and the liners being simply flat pieces, when we want to put one in we lay off the holes on the liner and drill it instead of the door. It is really quite an improvement.

C. M. S.

Oakville, Wash.

Use Hydraulic Check Valves

In a large manufacturing plant there are several high-pressure hydraulic pumps and presses. Since the pressure varies greatly and rapidly, especially in the pumps, it is difficult to maintain hydraulic gages on them. These gages soon become incorrect to such an extent that they are useless until repaired, owing to the rapid variations in pressure.

Is there a remedy to prevent the sudden pressure acting directly on the gages?

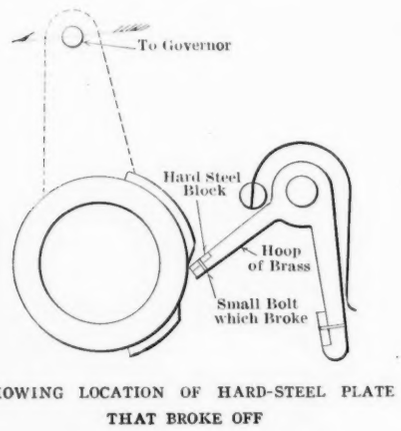
J. W.

Cleveland, O.

Quick Repair to Engine without Stopping It—Reclaiming Wiper Rags

We have two 1000-kilowatt units which are operated in parallel for feeding transmission lines, one of the prime movers being a turbine, and the other a cross-compound engine.

On the engine a few days ago the hard steel plate on the back of the hook that engages the governor knock-off cam fell off on to one of the high-pressure hooks, owing to the failure of a bolt. This allowed the engine to go to full stroke on one end. A piece of belting was run in in place of the steel plate, and held there, which made the engine cut off very well. Then a suitable bolt was obtained to hold the plate, one transmission line was thrown off, the throttle on the engine was shut and the rod unhooked, the engine being kept running all the while by current from the turbine-generator unit. The plate and bolt were slid into place and tightened, the rod hooked on and the throttle opened. The entire time consumed was a little less than two minutes, whereas if we had stopped the engine and then had to bring it up to speed again it would have taken at least ten minutes.



SHOWING LOCATION OF HARD-STEEL PLATE THAT BROKE OFF

In this plant rags are used instead of waste for wiping the machinery. To economize a little we take the best pieces about twice a week, when they are dirty and filled with oil, and cleanse them. We put them into a barrel, cover them with warm water, slice in about half a bar of laundry soap and add about one pint of gasolene; then we stir it all up and turn in steam. When the rags have been well boiled we fill up the barrel with water and allow it to overflow so as to carry off the oily stuff. This performance is repeated, and sometimes if the rags are very dirty we give them a third washing. It requires one man's time for about three hours, two bars of soap and half a gallon of gasolene. It saves about two-thirds of the rags and fits them for use over again.

Riverton, Ill. J. W. M.

"Advice on Pump Location"

Hoping that it may help John B. Ash, whose inquiry appeared in the January number, I submit the accompanying sketch of a temporary pumping outfit that was in use for a few weeks last summer.

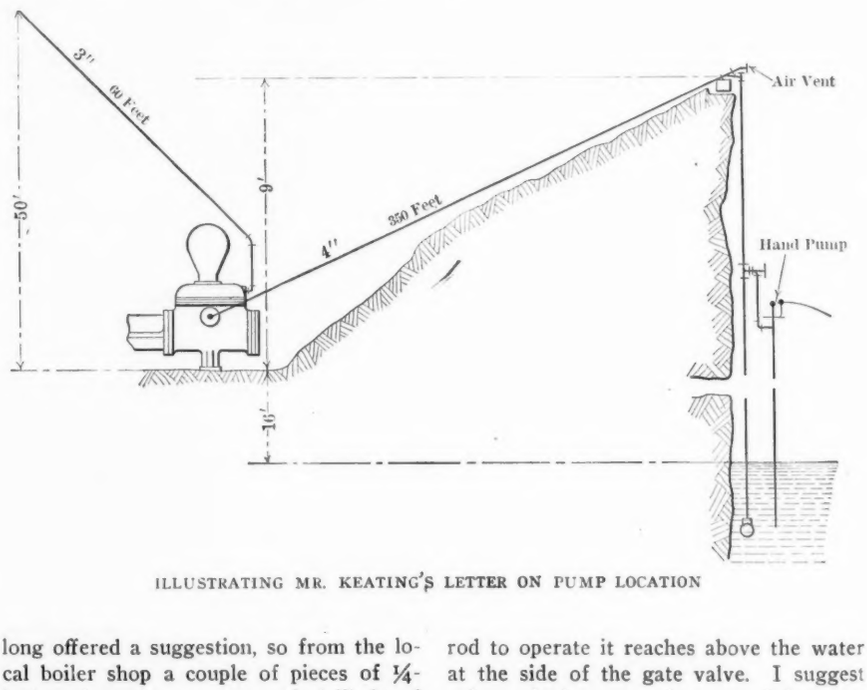
The pump was located about one thousand feet from the boilers and owing to the temporary character of the job no attempt was made to insulate the 2-inch steam pipe. Expansion was taken care of by a swing joint. The first attempt to remove the condensation was by placing a 1½x1½x3-inch bull-head tee in the horizontal pipe near the pump. A 3-inch nipple 12 inches long, capped and tapped for a ½-inch drip, constituted the drop leg and drain of the impromptu separator. This fixture proving itself useless as a separator, except when the pump was running slow, an 8-inch nipple 16 inches long offered a suggestion, so from the local boiler shop a couple of pieces of ¼-inch tank steel were procured, drilled and bolted to the ends of the 8-inch nipple which was set vertically near the pump. The steam pipe was let into the top through a 2x1½-inch reducing elbow with a screw nipple 1½ inches long extending downward about 9 inches, the steam to the pump being taken directly from the top. This arrangement furnished comparatively dry steam at all speeds; the condensation, roughly measured as it came from a small trap on the drain pipe, averaged about one hundred pounds per hour. In order to fill the suction pipe an ordinary well force pump with a 3-inch cylinder and 1¼-inch pipe connections was placed near the surface of the water and connected to the suction above the foot valve through an angle valve, as shown. The air valve on top of the vertical pipe was kept open until solid water was forced from it, after which, the angle and air valves being closed, the pump was started up and ran all day pumping about 150 gallons per minute. If Mr. Ash will be careful to have one central high point on the suction line and have this vented so that all the air in the pipe will rise to this point as the pipe fills, he will have no difficulty.

A foot valve, of course, must be placed on the end of the suction; if the foot valve remains tight the pipes should be full of water after periodical shutdowns.

I have a 6-inch centrifugal pump in the suction of which a 6-inch gate valve is placed just above the surface of the water. When the pump is stopped this valve is closed, and the pump is ready for work again, independently of the condition of the foot valve.

In freezing weather the water is drained from the pump through a 1-inch pipe tapped into the body of the foot valve. A rod to operate it reaches above the water at the side of the gate valve. I suggest using a duplex pump, because of the nearly continuous flow of water which it would produce in the long suction pipe. A standard 6x4x6, with a 3-inch suction and a 2-inch discharge, capacity about one-third of a gallon per stroke; or a 7x4x10, with a 4-inch suction and a 3-inch discharge, with a capacity of about half a gallon per stroke, would be right. A light service or tank pump could be used, but any of the standard sizes can be used almost anywhere else. I would have no hesitation in placing the pump at the low point indicated in Mr. Ash's profile sketch, and if the high point can be lowered easily, I would do that and insulate the steam pipe thoroughly. A prominent steam company claims to be able to conduct steam underground with a loss of less than five per cent. per mile.

Streator, Ill. P. J. KEATING.



ILLUSTRATING MR. KEATING'S LETTER ON PUMP LOCATION

The Passing of Joseph

Joe was the oiler at the Herringbone plant and while a steady, clever fellow, he was far from being an encyclopedia on mechanics. He had assisted Mr. Danvers, the chief, on various and sundry jobs and had acquired a general knowledge of local conditions as found in that plant, along with the idea that he was quite competent to handle things himself if the opportunity should offer.

Mr. Danvers had talked to him many times about the advantages to be gained from a well-studied course of correspondence instruction and had even told him to try a steam-engineering course and he would personally assist him in his studies. Joe finally succumbed and subscribed for the course, payments being made on the "easy" plan. Then his troubles began, and after a lapse of about eight months, the baby "fed the cat" with his second book in arithmetic, for which Joe duly gave thanks, as it gave him the needed excuse for dropping the whole thing.

Shortly after this Mr. Danvers was taken severely ill and had to resign his situation as chief, and as Joe was a pretty clever fellow, the manager promoted him to succeed to Mr. Danvers' position, but not to his salary, a twenty-five-cent raise being considered sufficient at that time.

Joe's first supply order contained a list of paints, and upon their arrival the floor of the engine-room was carefully laid off in checker-board squares and painted in indian red and brown, while at certain parts of the room cabalistic circles in light blue, containing brilliant red stars, kept the figure from becoming monotonous. As long as all this extra work was done by Joe, personally, and as he was keeping things going without any oiler to assist him, the manager evidently decided to overlook this artistic, if somewhat spectacular design in floor ornamentation.

Indicator cards were deposited regularly at the office, with the total horse-power written on each and the fact that Joe copied these results from an old card figured by Mr. Danvers long before, making slight changes each time so he wouldn't be detected by the manager, seemed to make no difference in his opinion of himself.

At one holiday season it was decided to clean one of the boilers, which, being a water-tube, took some time. The following day was Sunday, and early Monday morning the night fireman started the fire under the boiler to prepare it for the day's service. After a time he wandered down and took a look at the water and, seeing none, and noting that there was no steam pressure by gage, he went to the other end of the boiler room and started the feed pump. When he thought the water was about right, he again sauntered down to the boiler and again noted neither steam nor water. Thinking this was rather strange, he pulled open the furnace door and to his amazement found no fire, but lots of water.

Investigation disclosed four of the headers cracked, so he made no further attempts to start a fire under that boiler. The more complete investigation on the following morning also showed several tubes that were burned so badly as to necessitate removal. Further examination showed the blow-off valve to be wide open, in which condition it had been, supposedly, since the boiler was cleaned, without the fact having been discovered by any of the men in the boiler room during that time. Joe told the manager it was all spite work; that he had seen the boiler full and the blow-off shut, that someone had opened it to make trouble for him, because he had just been promoted, etc.

No clue being had to the spiteful one, the matter was dropped with the settlement by the boiler insurance company, they being the real losers in the trouble.

Shortly after this the management of the plant was changed and with the advent of the new manager, things assumed a different aspect for Joe. First a master mechanic was secured, who was known to be a capable man and then the whole mechanical end was placed in his charge, together with the responsibility for its successful operation. Joe was loud in his denunciation of this move. Had he not been chief ever since the departure of Mr. Danvers without a single kick coming from Mr. Bang, the former manager?

Being of this mind, as soon as the master mechanic arrived, Joe at once asserted himself. He cordially informed him that if there was anything he, the master mechanic, would like to know about that plant, that he, Joe, would show him, as he knew all about it. Furthermore, he, Joe, had received the offer of a very fine situation, but if the new man would raise his pay a dollar a day, he would stay with him long enough to get him started nicely, as he believed in using a new man right, etc.

The new man said he would think it over and two days later he brought another new man into the engine room to take charge and told Joe, if he cared to have a two weeks' notice, he would find him a job somewhere around the plant for that length of time, after which he would have to shift for himself.

Joe was not an exception in the great army of engineering employees, neither was he the rule, but he belonged to that class of men whose heads become abnormally large with the first application of the badge of authority. He did not see the true responsibility of the position and the future that awaited him if he proved successful; he did not see the need of straining every nerve to really make things better; he only saw the present glory of a position of authority and the immediate need of touching up the setting, so the occupant of the position might shine a little brighter among the men who were his former associates.

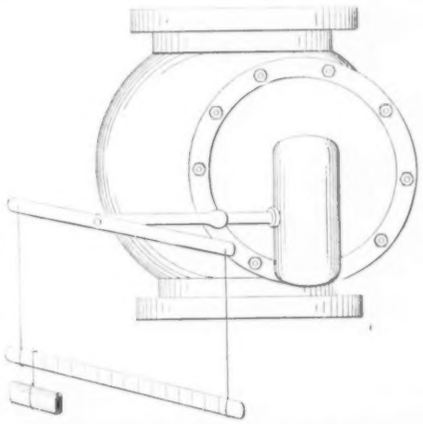
A proper pride in the position he holds is commendable in any young man, but when he thinks that he, and he only, is the main motive power in the plant and that without him, the plant would soon become a useless burden to its owners, his days of usefulness are over for a time at least and he is bound to get a tumble that will jar him badly, sometime when he least expects it.

L. L. ARNOLD.

East Greenwich, R. I.

Weighting a Back-pressure Valve

Mr. Wakeman's idea as to a convenient method of weighting a back-pressure valve, as explained by him in the November and March numbers, may suit the conditions in his case, but I think there is room for improvement. He would use



MR. SCHNEIDER'S SUGGESTION FOR WEIGHTING A BACK-PRESSURE VALVE

too many pulleys and ropes, while I do not think they should be used in connection with any kind of valve, because pulley ropes have a habit of becoming tangled when one is in a hurry. A better way would be to use wire, in connection with the arrangement shown in the accompanying sketch. By this means the valve can be adjusted to a nicety.

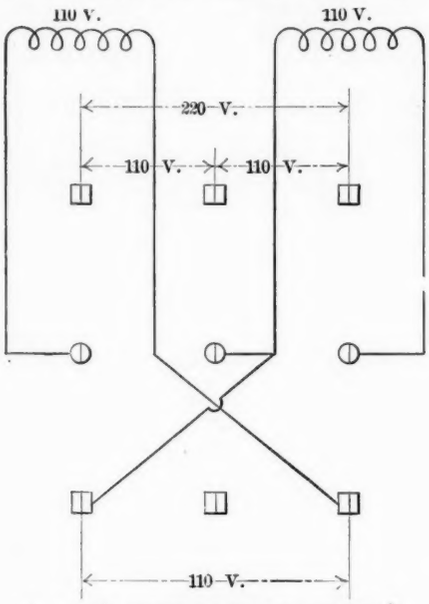
Mr. Wakeman also seems to have a horror of walking under a back-pressure valve weight. I suppose he takes as much pleasure in polishing the brass work around the engine as I do, and to do it he has to stand within range of the cylinder head 5 or 10 minutes each day, and yet he thinks nothing of it. Suppose the cylinder head should consider that it had as much right to take a day off as the back-pressure valve weight?

Incidentally, it might interest us Western pipers to learn how Mr. Wakeman managed to attach his 4-inch lever safety-valve on his boilers with levers 3½ or 4 feet long, and with 60- or 75-pound weights.

Streator, Ill. A. J. SCHNEIDER.

"A Problem in Transformer Coil Connections"

The acompanying diagram shows how I would solve E. S. Lincoln's transformer



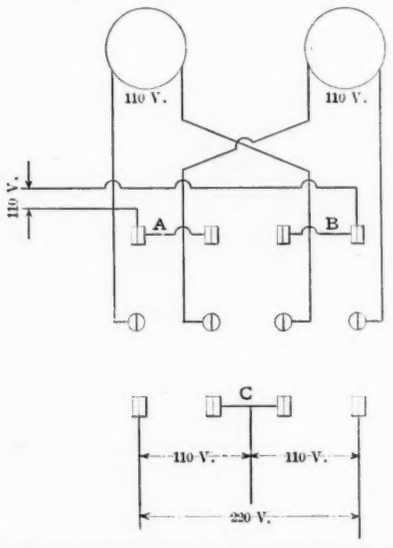
F. F. K.'S SOLUTION OF MR. LINCOLN'S PROBLEM

problem stated in March POWER, page 180, by means of one three-pole double-throw switch.

F. F. K.

Concord, N. H.

The accompanying sketch indicates the method I would employ to connect up the transformer in E. S. Lincoln's problem, on page 180 of the March number. The diagram shows the two coils, and the secondary leads of a commercial core-type transformer. The switch is of the four-



HOW MR. SWEGER SOLVES THE TRANSFORMER-COIL CONNECTION PROBLEM

pole, double-throw type. When the switch is up, the coils will be in parallel, which will give 110 volts; when the switch is down, the coils will be in series, which will give 220 volts. By taking a line from the junction of the two coils, a three-wire system could be obtained. Of course, it would be necessary to use "jumpers" on the switch at *A*, *B* and *C*.

CHARLES SWEGER.

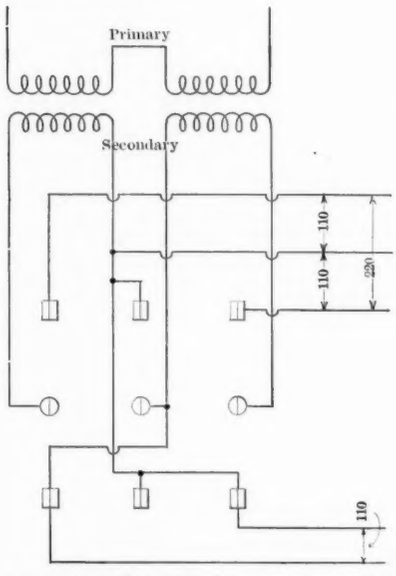
La Salle, Ill.

In reply to E. S. Lincoln's problem in the March number of POWER, page 180, I inclose a sketch of a method of connecting the two secondary coils, first in multiple to a 110-volt two-wire circuit, and second to a three-wire circuit. The switch is of the three-bar, double-throw type.

When supplying the two-wire circuit the center bar of the switch carries no current, as it is in no way connected to either the supply or receiving circuit. When in the opposite position the center bar joins the coils together and also connects them to the neutral wire.

A. C. BROWNE.

Dannemora, N. Y.



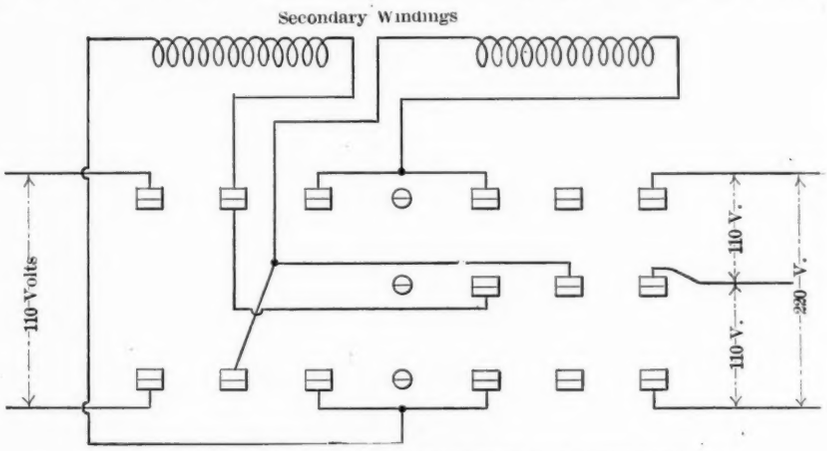
MR. MARTIN'S SOLUTION OF MR. LINCOLN'S PROBLEM

Herewith is a solution of Mr. Lincoln's problem in March POWER. A three-pole double-throw switch is used. The coils are connected in series when the switch is closed in the upper position, a neutral being taken off between the two coils. In the lower position the coils are connected in parallel; one of the lower jaws is not used and may be removed, if preferable.

GEORGE W. MARTIN.

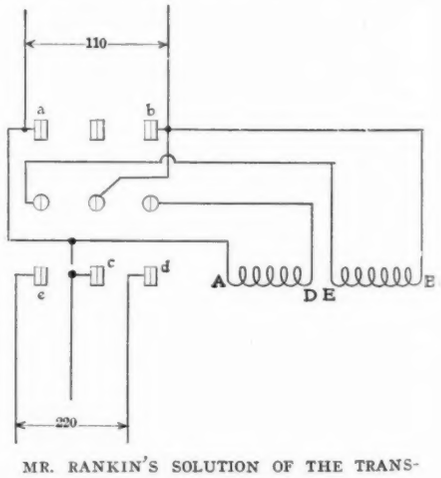
Brooklyn, N. Y.

The accompanying diagram shows one method of solving Mr. Lincoln's problem in transformer-coil connections which was published in the March number. A double-throw, triple-pole, knife-blade switch is



A. C. BROWNE'S METHOD OF CONNECTING THE SECONDARY COILS

used, and three of its jaws, *a*, *b* and *c*, are connected permanently to two of the coil terminals. When the switch blades are thrown upward, the coil terminal *c* is connected to the circuit terminal *a* in parallel with the other coil terminal *A*, which is permanently connected to that jaw. The coil terminal *D* is similarly connected to the circuit terminal *b* and the coil terminal *B*, so that the two secondary coils are in parallel on the 110-volt circuit. When the blades are thrown downward, the coil terminal *E* is connected to the circuit terminal *e*, the coil terminal *B* to the circuit terminal *c*, to which the terminal *A* is permanently connected; and the fourth terminal *D* is connected to the circuit terminal *d*. This puts the two coils in series, with the neutral circuit



MR. RANKIN'S SOLUTION OF THE TRANSFORMER-CONNECTIONS PROBLEM

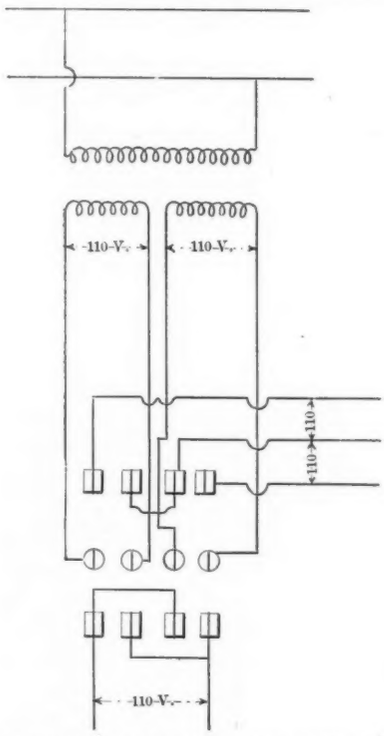
conductor connected to their junction point.

CHAS. O. RANKIN.

Craftonville, Cal.

[Mr. Rankin's solution, while operative where the 110-volt circuit is never connected to any other source of current than the transformer shown in the diagram, would not be permissible otherwise, because the 110-volt circuit is short-circuited when the switch is closed downward.—EDITORS.]

In answer to E. S. Lincoln's problem in transformer-coil connections, on page 180 of the March number, I would use a four-



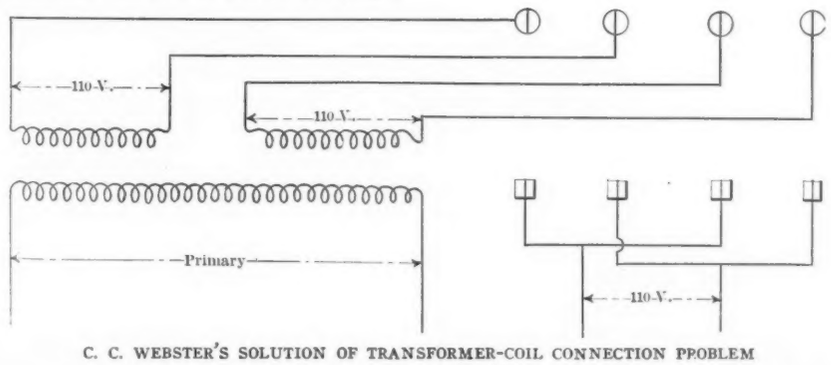
MR. BRAND'S SOLUTION OF MR. LINCOLN'S PROBLEM

blade knife switch and connect as indicated in the accompanying sketch.

H. B. BRAND.

Brooklyn, N. Y.

Mr. Lincoln's problem in transformer-coil connections can be best solved by using a four-pole, double-throw switch connected up as shown in the diagram herewith.

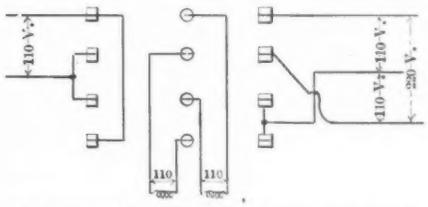


C. C. WEBSTER'S SOLUTION OF TRANSFORMER-COIL CONNECTION PROBLEM

C. C. WEBSTER.

Schenectady, N. Y.

The diagram herewith is a solution of Mr. Lincoln's problem in transformer-coil



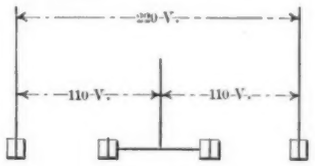
MR. CERNY'S SOLUTION OF MR. LINCOLN'S PROBLEM

connections. One two-throw four-pole switch will do it.

FRANK W. CERNY.

Mesa, Ariz.

The accompanying diagram of connections shows my method of solving the problem on page 180 of the March number. The single switch *S* is of the triple-pole, double-throw type, and when in the



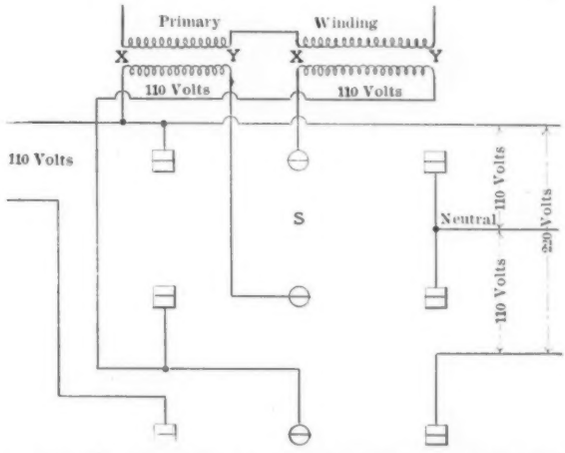
right-hand position it will connect the two secondary coils in parallel to one lighting circuit. Throwing the switch to the left puts the coils in series and connects them to the opposite lighting circuit. The neutral wire of the latter circuit is connected to the middle point of the two coils when in series.

E. S. LINCOLN.

Boston, Mass.

"A Case of Demagnetized Field Magnet"

In regard to Mr. Doughten's case of demagnetized field, one is led to believe that it was caused by throwing off the load before stopping the engine. If this be true, I can come to only one conclusion, and that is that it was due to self-induction caused by opening the switches. It is well known that when a circuit is



MR. LINCOLN'S SOLUTION OF HIS TRANSFORMER-COIL CONNECTIONS

opened in which there is inductance there is a momentary rush of the current in the opposite direction to that in which it was flowing before the circuit was opened, and this current will change the polarity of the magnet it affects. If strong enough it results in a reversal of the machine, but in this case it seems it was strong enough merely to neutralize the residual magnetism. If the motors were running—and I presume they were, for he speaks of this generator as operating motors—the circuit was a highly inductive one and I can readily conceive this to be the cause of the trouble.

FRANK W. CERNY.

Mesa, Ariz.

When first I read the letter and question of Mr. Doughten on the above-mentioned subject, and the appended editorial note, I wondered why so simple a question had been referred to POWER readers for analysis; but when I tried to think out an answer, I was brought up with a sudden jolt to the realization that I must go deeper. Then I began to look up references on this subject, but they all told me what I already knew, viz.: the simple statement that when the flow of current was stopped in the windings of a field magnet, the magnetic lines of force were

reduced to a very slight value, which was called residual magnetism; that the loss of residual magnetism could be overcome by connecting several batteries to the coils, and thus induce a flow of current. These were the only direct statements made in the hand-books which I have.

Elsewhere I find what seems to be the point at issue, namely, that the residual magnetism of a closed magnetic circuit will remain a long time if the circuit is unbroken; but when once broken, then the said magnet or circuit must be remagnetized by the application of an electric current.

This may apply in Mr. Doughten's case, as his trouble occurred only when the engineer opened the main circuit before shutting off the engine. On the other hand I have done this very thing hundreds of times on a compound-wound machine with no ill effects.

Mr. Doughten fails to say what kind of a machine his was, whether shunt, series, or compound, and this again might make a difference.

I shall note with interest the replies as printed and trust that pertinent references, if found, will be given in full, with the authority for the same.

L. L. ARNOLD.

East Greenwich, R. I.

My theory regarding the destruction of the residual magnetism in W. S. Doughten's dynamo is this: When the engineer pulled the switches the circuit supplying exciting current to the dynamo field magnet was opened suddenly, causing a reversal in the direction of the magnetic flux in the field-magnet cores and yoke, and this had a tendency to destroy the residual magnetism in them. In that case the machine would not build up until a current was supplied to the field magnets from some external source.

INTERESTED.

Junction, Ariz.

In regard to Mr. Doughten's "case of demagnetized field magnet" I think that when the dynamo load was thrown off an induced current of great volume and high voltage was set up, flowing through the armature in the same direction as the original current. Although this induced current flowed but for an instant, it was sufficient to cause a high armature reaction that "killed" the fields. The resistance of the rheostat being "all in" helped to bring about the result.

H. B. BRAND.

Brooklyn, N. Y.

In reply to Wm S. Doughten's letter on page 250 of April POWER, relating to the failure of a dynamo to "pick up," I should judge that the loss of residual magnetism by his generator was due to the demagnetizing effect of back ampere-turns on the armature, the action of which can be understood from the accompanying diagrams. If the field is excited and no current is flowing through the armature, the magnetism or lines of force would flow in a straight line through the armature, as indicated by *F*, Fig. 1. When current flows through the armature, it sets up magnetic lines of force at right angles to the field magnetism, which causes distortion of the latter magnetism, and the field lines of force will flow as shown by *F*, Fig. 2.

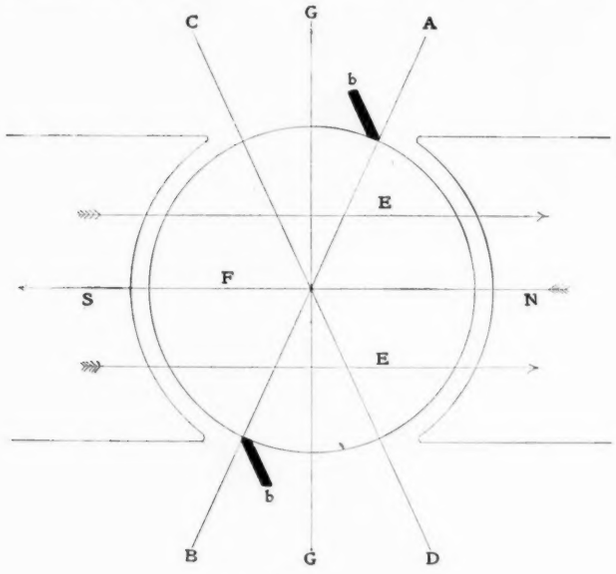


FIG. 1

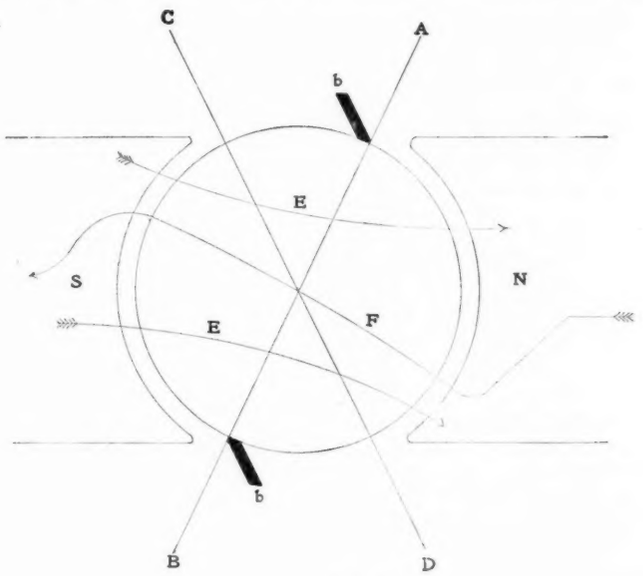


FIG. 2

This distortion depends on the strength of the field magnet and the current flowing through the armature; the stronger the field magnet the less the distortion; the larger the armature current the greater the distortion.

In order to secure sparkless commutation it is necessary to shift the brushes *b b* forward to the neutral line *A—B*, Fig. 2. The armature conductors between the lines *A—C* and *B—D*, which are governed by the position of the brushes and the amperes flowing through them, constitute the "back" ampere-turns; that is, they exert a magnetizing influence in an opposite direction to the field magnetism, and therefore tend to destroy it. See *E E*, Fig. 2.

The armature conductors between the lines *A—D* and *C—B*, Fig. 2, and the amperes flowing through them, constitute the "cross" ampere-turns that produce distortion. It will also be seen that they slightly neutralize the effect of the "back" or demagnetizing ampere-turns.

In Mr. Doughten's generator, when the load is on, the field flux is distorted by the armature current. The brushes are shifted forward to prevent sparking and the condition is that shown in Fig. 2. Therefore, when the load is left on and the engine is shut down the distortion is approximately the same until the generator ceases to deliver current, as the current and the magnetism die out together; consequently, the field magnet does not lose its residual magnetism. But suppose that the generator is in operation with the load on, the same as before (Fig. 2), and the main switch is opened. The condition changes to that in Fig. 1, in which there is approximately no distortion, as the only current through the armature is that required to excite the field magnet, which however small is sufficient to produce the back ampere-turns that produce a magnetic stress in the opposite direction to that of the field magnet, as along the lines *E E*, Fig. 1, and demagnetizes the field magnet to such a low value that the generator will not "pick up" when started again.

It will be seen from the diagrams that if the brushes are shifted back the demagnetizing ampere-turns are eliminated, which would occur when the lines *A—B* and *C—D* were brought to the position of *G*, Fig. 1. If Mr. Doughten will shift his brushes back to the theoretical neutral point when he opens the main switch and shuts down his engine, and will allow the generator to come to rest with the

brushes in that position, I think he will find it ready to pick up when started again. While it might not be practicable to shift the brushes every time the generator is stopped and started, I believe it would prove interesting to readers of POWER if he will act on the above suggestion and report the result.

G. E. RALSTIN.

Indianapolis, Ind.

"Piping a Steam Pump and an Injector"

Regarding Mr. Wakeman's proposed arrangement, shown in Fig. 2 on page 31 of the January number, I can just imagine what an engineer or fireman would say after he had been obliged to climb a ladder several times to operate one of those high valves *7* and *8*. I have often noted that Mr. Wakeman is over addicted to the use of angle valves, however. To be sure, it is no wonder that objections are found to the piping as shown in Fig. 1, page 30, the arrangement to which Mr. Wakeman also finds objections.

Letting alone the inconvenient placing of the valves, there is more piping than is needed. For instance, the piping running from the suction pipe of the injector to the boiler feed-pipe could be eliminated, and not be missed, by putting a by-pass around the injector as shown at *A* in the modified plan Fig. 1 herewith. If friction is of so much account that the difference between two kinds of valves is of moment, then the friction of 15 or 20 feet of useless piping would be noticeable.

Another piping arrangement which would improve matters would be to connect the suction pipe of the pump, as shown at *B* Fig. 1. The pipe and valve over the water pail do not seem at all necessary. If it were absolutely required to have a bleeder pipe it could be connected to the same fitting and extended downward as shown at *C*, Fig. 1. It would look better there and be out of the way.

An objection to the angle valves in Mr. Wakeman's plan is that they have to be placed beyond the reach of the attendant. Frequently, if the stem is to be freed from the pressure of the steam or water, the valve would have to be set with the stem vertical, which makes it difficult to pack, and still more so if it is so located that the attendant is obliged to work on a ladder. If the matter of pressure against the valve stem is not considered, then it will not make any difference which way

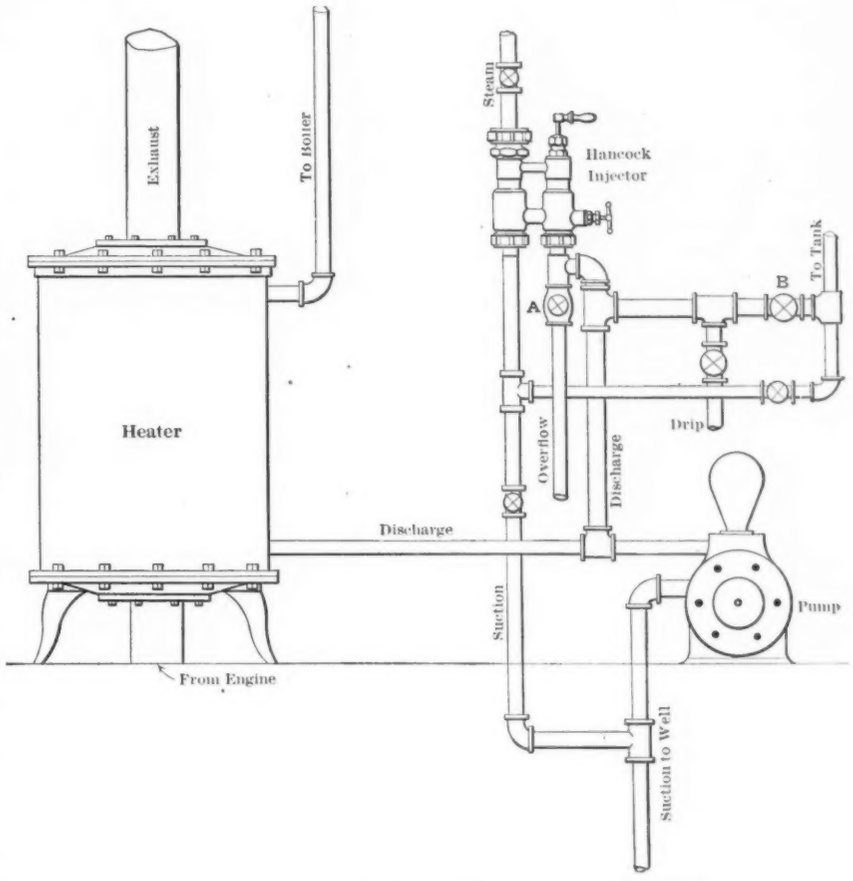


FIG. 2. SIMPLE ARRANGEMENT OF PUMP AND INJECTOR PIPING

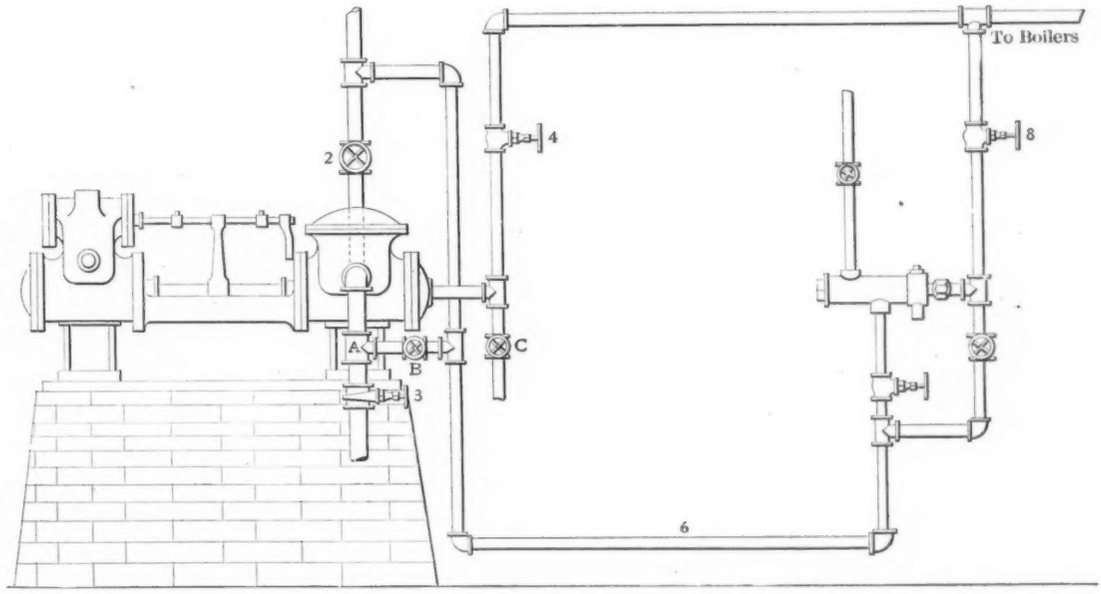


FIG. 1. MODIFICATION OF THE PIPING ARRANGEMENT TO WHICH MR. WAKEMAN OBJECTED

the valve is placed in the pipe. I have failed to note that angle valves have become very popular in any of the steam plants I have had the pleasure of visiting.

Speaking of injector piping reminds me of a steam plant which I operated some years ago. I think the piping was as simple as it could be for the pump and the injector. Fig. 2 shows the arrangement. It will be seen that the pump suction pipe leads to a well; the discharge pipe from the heater is tapped to the pump suction to save running an extra length of piping to the well, and the discharge from the injector is tapped to the discharge from the pump, instead of running to the heater in a separate line.

Another feature was the manner of making the discharge pipe answer for a cooling pipe in case the injector became heated. The overflow is tapped to a drip-pipe running to the outside of the engine room, and an extension connected to a water tank on the top of the building. If the injector became heated, the valve *A* was closed and the valve *B* opened, allowing the water to run from the tank piping and thus cool the injector. It may be thought that it would be impossible to operate both the injector and the pump at the same time, but the trick has been done time and time again. This feature came in very handy more than once when the time for shutting down came and there was not sufficient water in the boiler for the night.

Another objection that I have seen raised to piping the injector to a heater is, that the heater will reduce rather than raise the temperature of the feed-water. This may be in some instances, but not in the majority of steam plants. A good heater will heat the feed-water from 200 to 212 degrees, but I have yet to see the injector that will do so.

INJECTOR.

Cleveland, O.

"Should Lignite Be Wet or Burned Dry?"

The inquiry by John MacGregor on page 259 of the April number, concerning lignite, brings up an interesting point, but it is one that Mr. MacGregor should be in a better position to determine than others who have no chance to experiment with the steaming qualities of brown lignite. As San Antonio is located in the brown-lignite field, I assume that this is the coal he uses and to which he refers. We all know that wetting coal is a fixed habit with some firemen, and it is the writer's experience that many of these firemen can give no practical reason for doing so. Their excuses are numerous, some of them mere fallacies.

It would seem that the less moisture that is introduced into the furnace, the fewer heat units should be required for its evaporation, as evaporated it must be before the coal can be burned; therefore, if the coal is dry these heat units would be expended on evaporation inside of the boiler instead of outside. As the moisture content of Texas lignites runs very high when fresh from the mine, in many cases averaging 33 per cent., it would seem as if there were already moisture enough. On the other hand, the Government report from the St. Louis coal-testing plant contains this statement: "The points of greatest weakness are our ignorance of the character of the volatile combustible matter and also of the form or nature of what is usually called moisture. A few experiments made at St. Louis seem to indicate that something more than moisture frequently passes off of coal at ordinary temperatures and hence the moisture as shown in the anaylses probably includes some of the lighter hydrogen compounds.

"Also, it is doubtful whether we should regard water as a non-fuel element for in many cases it is dissociated and its hydrogen becomes available for fuel purposes."

A. G. Leonard, in a paper on cretaceous lignite, gives it an average moisture content of 37.92 per cent. when fresh from the mine, and adds: "When exposed to the air for a short time the lignite loses much of its moisture and crumbles into small fragments. For this reason it is very desirable that the coal be used as promptly as possible after it is mined, before it has time to disintegrate, or 'slack'."

This would seem to indicate that the fireman who uses the water is in the right, providing the water he applies restores the lignite to something like its original condition. Despite the above assertions, however, the heat balance on the test of Texas lignite at St. Louis credits a loss of 7.3 per cent. to moisture in coal and moisture formed by the burning of hydrogen. After considering the above we might well ask, "Where are we at?"

I would suggest that Mr. MacGregor make alternate tests with wet and dry lignite, keeping a careful record of the results, and give to the readers of POWER the data in detail.

East Greenwich, R. I. L. L. ARNOLD.

Why the Gage Glass Did Not Show the Water Level

During a long experience as erecting engineer I have seen some queer piping jobs, but none more ridiculous, it seems to me, than that illustrated in Fig. 1, which shows the water-column connections on one of three horizontal tubular boilers which were all piped alike. Complaint had been made that it was impossible either to obtain or maintain a water level in the gage glass, so I was sent to straighten the matter out.

I found that the 1½-inch feeder from the feed-pump was connected, by means of a three-way elbow, both to the water column and the bottom of the boiler, but the connection between the elbow and the boiler was 1 inch in diameter instead of 1½ inches. There was no connection between the water column and the steam space in the boiler; instead, the feed pipe led from the top of the water column across the top of the boiler, horizontally, and down the left side to a point near the bottom, where it entered the boiler. The water column and gage glass were constantly filled with feed-water, and consequently the water level in the boiler could not be shown in the gage glass.

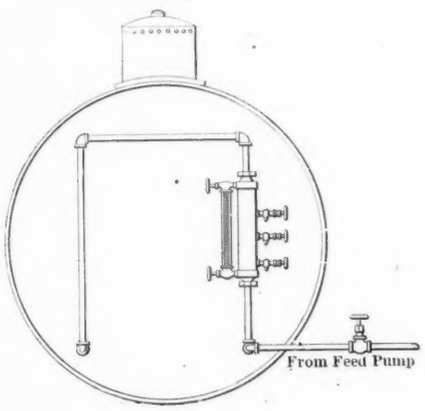


FIG. 1

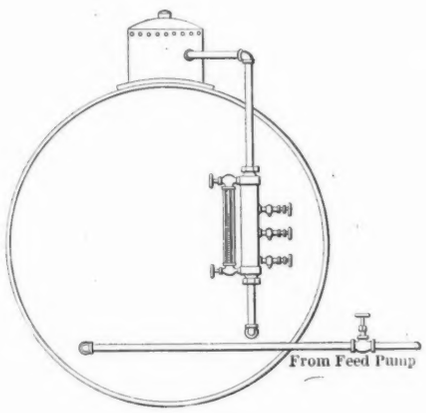


FIG. 2

It did not take long to devise the arrangement shown in Fig. 2, whereby the feed was led directly into the bottom of the boiler, connection was made between the water column and the steam drum, and the lower end of the water column was connected to the boiler, as shown. These changes made, the gage glass performed its functions readily.

I told the engineer in charge that I did not understand how anybody could expect proper results from the old arrangement. He replied that he had been in the business 25 years and couldn't "see anything the matter with it." I answered that it was beyond my comprehension why he wasn't in heaven years ago. Then I left without waiting to learn whether he grasped the significance of my criticism.

P. E. JAY, JR.

New York City.

Why It Pays to Employ a Consulting Engineer

I was much interested by an editorial in December POWER which treated of "Electric Generator Drive," with reference to the employment of a consulting engineer. I believe that there is never a case where it is not financially practicable to employ a good consulting engineer, for whatever you spend without his aid may be largely wasted through the lack of knowledge of the proper way to install or use machinery. Repeated cases have shown me the lack of wisdom in depending on the engineering staffs of large companies, not because of their lack of engineering ability—far from it; but simply because they are in business to manufacture and sell their machinery, and in the competition between manufacturers no two of them agree on the methods to be used, or the machinery to be installed, because each wishes to be different, and also to do the work more cheaply.

As a fact, the real differences may be small, but to the uninitiated purchaser these differences may look large, and when he examines the various suggestions he is all at sea. Also, the engineer of a big company takes the prospective customer's measure, so to speak, decides how much he is likely to spend, and makes a layout according to the conditions as he "sizes them up." No two men size up the situation alike, and here again a variation is likely to occur. All of them will undoubtedly try, and will improve, the economy of operation, but not to the full extent possible. In connection with this attention should be called to the fact that the engineer of the manufacturing company will report only on the character of the work in which his company may be engaged. As, for instance, the engineer of an electric company, called in to look over a plant to improve the drives, will report favorably for motors, in greater or less number, but he will not look further to see if other methods may not be better.

These remarks are not fancy, but facts. I know of a number of cases which justify my statements. Among them: A plant was examined by representatives of several manufacturing companies, who sent in their suggestions. All advocated motor drives, but to the dismay of the customer the number of motors varied from six to thirty-eight. Do you wonder that he decided to do nothing? In another case, alternating current being used, individual motors were placed on small sewing machines, resulting in very poor economy. Another case which came directly under my notice was the recommendation of three motors, where it was proved that nine were far away better, both in economy of operation and first cost, on account of the reduction of interconnecting belts, shafts, etc.

The consulting engineer is not bound by any ties to recommend a special method of driving or special make of apparatus. He is called in to improve the operating conditions of the plant, and will report for that method of driving which will best suit the needs of the plant. This point alone is worth paying a good price for.

The manufacturer has little to lose in his dealings and can afford to give his services free, if he can sell his machinery. The consulting engineer, on the other hand, has a great deal to lose and to gain. His reputation suffers if he does not make good, and it is enhanced if he does. Therefore, he puts all his energy into making the plant as efficient as possible, and at *the lowest possible cost,* since the interest on the investment bears largely on the expense of operation. Consultation by mail is sometimes possible for general results, but for the final plans, etc., the only method for getting accurate results in installation and operation is a personal visit to the plant and a study of the conditions.

Another point worth noting is that the engineer of the manufacturing company will give only general details, and will not pay the needed attention to petty matters. There are many cases in which alternating-current machinery has been installed, and where trouble has been experienced, due very largely to inaccurate or incomplete information regarding details. Among these mistakes are the inclosing of the separate wires of a three-phase system in iron pipes; poor feeder layouts, giving rise to heavy inductive effects and the like. A consulting engineer will look after these details and avoid such troubles, as well as many others.

In conclusion, I think I am warranted in saying that the money spent for a good engineer is a guarantee of good results, and is more than saved in the service rendered.

Boston, Mass. HENRY D. JACKSON.

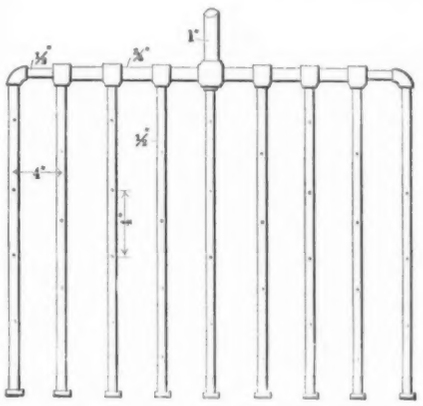
Shut-down Avoided by Handy Makeshift—Double versus Single Piston Packing Rings

On page 735 of December POWER is a short article giving an account of repairs on a brick chimney, the top of which had been damaged by lightning. The article states the works were shut down for seven days, the time that it took to repair the chimney.

During the construction of the Albany and Hudson electric railway word came to me at Valatie, at about 11 a.m. one day, that the stack of our Rensselaer lighting station had been blown down the night before, and a new one must be put up so that the plant could be run that night. I arrived on the scene at 2:30 p.m., and found that the iron stack had been broken off about ten feet above the roof. I enlisted the aid of the steam fitter, and made up a steam blower, similar to that shown by the accompanying sketch, the vertical pipes being ½ inch in diameter, four inches apart; the holes were 1/16-inch, four inches apart. This was applied to the boiler, and the station was started up in time. This blower continued to furnish blast for the boiler, with the shortened stack, for six months, when this station was shut down, as the company's new station was then ready for operation.

I now think it would be well to have the ½-inch pipes of the blower five inches apart, the holes being spaced as before. But above 100 pounds steam pressure I would make the holes 3/64-inch, as the object is to get a large volume of air with a small amount of steam.

That there has been considerable trouble caused by the wearing down in the low-pressure cylinder of compound engines is well known. The Watts-Campbell



MR. CRANE'S STEAM BLOWER

Company had an engine of this type that gave the usual trouble when starting up, and they gave the matter a pretty thorough investigation. They took off the back cylinder head and ran the engine single-acting on the low-pressure side. They also procured a squirt gun with which to inject oil into the cylinder by hand. In spite of the fact that the hand feed was applied where it seemed to be most needed, supplemented with automatic feed of the regular type, the bottom of the cylinder would get so hot that at times it was apparently afire. The trouble appeared to be in the double-ring piston packing, the head ring apparently wiping off the oil. One ring was therefore taken out and the trouble stopped.

The use of two rings next to the outside of the piston reduces clearance, as well as providing a reserve ring in case one happens to break, but efficient oiling is more sure where there is but one ring, even when the edges of the ring are rounded over, although the assumption is that when so rounded the ring rides up into the oil.

Waterbury, Conn. W. E. CRANE.

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The Turbine at Sea

David Johnston, chief engineer of the Allan Line, the first to use turbines in the transatlantic service, has been quoted as saying that the next turbines he bought would "have connecting-rods on them." In view of Mr. Johnston's previous favor for the turbine this intention to return to the reciprocating type has occasioned some comment, but it is not necessarily a broad condemnation of the newer type of motor. The steamers in which the turbines were used are in the Toronto trade and have to make a long river trip at reduced speed. It has been understood that the turbine ship "Carmania" of the Cunard line used more coal than her sister ship the "Caronia," fitted with reciprocating engines; but it is now said that since the turbines have been run at the full speed for which they were designed she is burning from five to eight per cent. less than the other vessel and saving considerable in the engineer's staff on account of the smaller number of oilers required.

In connection with the effect of speed on turbine efficiency we are told that the turbine-fitted Channel steamers which are obliged to leave and arrive at stated times can save coal by lengthening their course and maintaining full speed rather than by going more directly between their ports at the slower rate required to meet their schedule.

Bread from Brick-bats

If somebody should announce that he had discovered a substance which, sprinkled upon a peck of builders' sand, would change it into a dozen cut-glass tumblers, we wonder if the daily papers would proclaim the discovery in all seriousness and print two-column pictures of the discoverer, as they are doing of John Elmore, the alleged discoverer of a compound which, sprinkled upon ashes, "causes them to burn with a fiercer heat than the best bituminous."

John is "so illiterate," one of the articles states, "that he is compelled to seek assistance in writing to drug houses for his ingredients"; but in our opinion John knows a good deal more than the newspaper man who prints John's portrait on the front page of a metropolitan daily, with a two-line title, turn-over story full of such rot as: "He has without doubt or question created a chemcial compound which is bound to have a tremendous effect on coal production so long as coal is used as fuel." "He has not even an elementary knowledge of organic chemistry, and yet he has discovered a combination which the most eminent chemists in the world have been seeking for years." We say "knows more," because it could not be that the publishers of such papers, knowing better, would let such misleading twaddle appear in their reading columns even for money.

Of course, the man has not invented nor found anything of the kind. Ashes, except for the modicum of unburned coal unwittingly removed with them, are absolutely incombustible. If John has found anything outside of carbon or hydrogen or some of their compounds which will burn, he might as well use sand or anything else for a wick as ashes, and save buying the initial coal. The announcement that such a substance, commercially available, had been discovered, would be equivalent to the announcement of the discovery of another continent on this much explored sphere.

The only harm that comes from such articles as these, aside from making their promoters and publishers ridiculous, is the divorcing of still more ignorant and trusting dupes from their good money through the apparent indorsement which the alleged discovery receives from such a presentation.

Effect of Overloading an Engine

Why should an engine break down with an overload? In a properly proportioned and adjusted automatic cut-off engine the maximum pressure is admitted at the commencement of the stroke and with the highest load. The pressure is no greater and probably not so great at quarter stroke as at tenth stroke; at half stroke as at quarter. Why, then, if an engine will stand up at all under a given pressure is it to be feared that it will experience distress when carrying an overload—when carrying no greater pressure but carrying the same pressure a little longer?

When the piston is at the commencement of its stroke and the crank is on the center, the pressure on the piston has no effect in pushing the crosshead down upon the guide. As the piston moves forward and the connecting-rod begins to point upward, the engine running "over," it thrusts the crosshead down upon the guide and the more the rod points upward the harder it pushes downward. It will point upward most, i.e., it will have its greatest angularity when the crank is perpendicular, which will be at something over half stroke on the head end and at as much under half stroke on the crank end. If the maximum pressure continues up to this point the guides will be subjected to greater stresses than they would be with an earlier cut-off.

But an engine usually fails under an overload not by breaking down but by heating up. The heat generated in the surfaces is proportional to the pressure between the surfaces and the distance through which they move under that pressure. With a light load the pressure on the guides and bearings is relieved

early in the stroke, and heat is generated no faster than the bearing can dissipate it; whereas, with a heavy load the pressure, although no greater, is applied longer generating more heat and often generating it faster than it can be radiated, so that the heat accumulates and a hot bearing is the result.

The Gas Engine's Waste Heat

It is more than probable that the heat rejected in the exhaust gases and jacket-water discharge of gas engines is not utilized to anything like the extent it might be, but it is also true that it is not utilizable to the extent commonly supposed. The most practical way to apply the heat of hot gases as a rule is to use water as an intermediary, and the natural idea is to use the jacket-water discharge so as to get the advantage of its heat, and add to that the heat of the exhaust gases, or such portion of it as can be utilized without boiling the water. There would be no advantage in attempting to evaporate the water into steam unless the steam could be actually worked back into water, because the whole of the available heat would evaporate only a small part of the jacket discharge water and the latent heat in the steam would be lost. But all of the jacket discharge water and a little more could be heated to 210 degrees. The question is: Of how much practical value would it be?

For example, suppose a 100-horse-power engine to be operating at full brake load on 9231 cubic feet of producer gas per hour, the gas having a calorific value of 130 B.t.u. per cubic foot. This would be 12,000 B.t.u. per hour per brake horse-power, which is a quite common rate. The heat distribution for such an engine would ordinarily be 500,000 B.t.u. per hour carried out in the exhaust gases, 400,000 in the jacket discharge water and 45,500 B.t.u. lost in friction and radiation. Allowing 40 pounds of cooling water per hour per brake horse-power, which is a practical figure, there would be passed through the jackets 4000 pounds of water per hour and the temperature would be raised 100 degrees Fahrenheit; if the initial temperature were 60 degrees the temperature at discharge would obviously be 160 degrees. By putting the water through a small heater similar to a closed boiler feed-water heater its temperature could be readily elevated to 210 degrees by taking out only 200,000 of the 500,000 B.t.u. carried out hourly in the exhaust gases. Supposing the explosive mixture in the engine cylinder to be composed of 1.1 cubic feet of air for each cubic foot of gas, the total volume passed through the inlet valve per hour would be 19,385 cubic feet, which would doubtless expand to 28,000 cubic feet when exhausted. If the engine speed be 160 revolutions per minute there will be 80 explosions per minute, or 4800 per hour, so that each exhaust delivery will be $28{,}000 \div 4800 = 5.83$ cubic feet, and the heater would need to have at least that capacity, or say six cubic feet.

So far, so good. With a 100-horse-power engine running at full load, and a heater of six cubic feet gas capacity, we are able to make available 600,000 B.t.u. per hour, stored in water at a temperature of 210 degrees. This would be sufficient to warm a workshop of moderate size in ordinary winter weather, or useful for any manufacturing operations requiring such a moderate temperature as 200 degrees or under, but the conditions must obviously be especially favorable in order to turn to account the waste heat. The available quantity of water per hour (4000 pounds, or 480 gallons) is a mere bagatelle in most cases where hot water is used industrially. Moreover, these figures are based on full-load operation, which usually covers only a small part of each day.

One of the persistently alluring possibilities is the evaporation of part of the jacket water into steam and the use of the steam in a cylinder working on the same shaft with the gas-engine cylinder. This plan figures out rather poorly thermodynamically, but it has such operating attractiveness that it is really difficult to cast it aside. The steam cylinder would, of course, add just the characteristic that the gas engine lacks, namely, reserve capacity; and the preferable method of operation would be to work the steam cylinder normally at its most economical load and allow the gas cylinder to take care of fluctuations below full load. Unfortunately, on the basis of the figures already cited, and which may be considered representative, the heat available in the exhaust gases would evaporate only 375 pounds of the jacket water per hour into steam at 90 pounds gage pressure, assuming the boiler to have about 80 per cent. efficiency. This means, obviously, that the steam cylinder would develop only about 12 horse-power. Now, a gas engine of 120 horse-power, running with a load of 112 horse-power (the combined output of the previously assumed gas and steam cylinders) would require about 113 pounds of coal to be burned per hour in the producer, while the composite unit would require 100 pounds. Would the saving of thirteen pounds of coal an hour during the full-load period of each day, and the smaller saving during the remainder of the daily run, pay the fixed charges on the difference in the first cost of the two equipments, not to mention the increased consumption of lubricating oil by the composite unit? We think not. But we wish somebody would find a profitable way to utilize the available part of that waste heat.

The Direct-current and Induction Motors

The direct-current shunt-wound motor and the alternating-current induction motor have many operating characteristics that are alike in kind, but some of them differ in degree. Both motors require provision against a heavy rush of current through them when first connected to the supply circuit. The armature of the direct-current motor is so low in resistance, however, that it constitutes practically a short-circuit when at standstill. The induction motor winding is also of rather low resistance, but it contains so many convolutions and the magnetic leakage is such that when connected to an alternating-current circuit its inductance keeps the rush of current considerably below what it would be with direct current. Because of this, a comparatively large induction motor may be thrown in circuit without damage to itself. It does take a heavy "gulp" of current for a moment, but nothing like the rush that would go through a direct-current motor of equal output.

When in operation, both motors drop slightly in speed as the load increases, and *vice versa*. This is due to precisely the same cause, namely, the need for more current in the armature, and the necessity of reducing the counter electromotive force in order to get it. Variations in the impressed voltage also affect the torque of both types of motor, but in different degree. The torque of an induction motor will vary as the square of the voltage applied to its terminals, while the torque of a direct-current motor is affected very slightly by ordinary variations of impressed voltage. This difference is due to the fact that the iron in the field magnet of the direct-current motor is so highly magnetized that the field strength does not change proportionally with changes in the magnetizing current, while the reverse is true of the induction motor. The torque of both machines is proportional to the field strength and the armature current. Reducing by 10 per cent. the voltage at the terminals of an induction motor will reduce its magnetic flux 10 per cent., and this will reduce its rotor current 10 per cent.; hence, the torque will be reduced 19 per cent. With a direct-current machine, the 10 per cent. reduction in impressed voltage would reduce the field strength much less than 10 per cent.—usually 3 or 4 per cent.—and this would reduce the counter electromotive force and allow *more* armature current to pass, thereby tending to keep up the torque. Hence, when carrying a heavy load, the induction motor is much more sensitive to changes in voltage than the shunt-wound direct-current machine.

A New Fuel for Internal-combustion Motors

By Howard Greene

The propagation of flame in a gas consisting of vaporized alcohol and sufficient air to insure complete combustion of the hydrocarbon is so much slower than in a gas in which the hydrocarbon is gasolene that a motor designed to run on gasolene will perform very inefficiently on alcohol. Theoretically the ideal condition would be complete combustion at minimum volume—highest compression—and only expansion during the piston stroke. While practical considerations modify this statement, it is necessary to obtain complete combustion early in the stroke in order to get high thermal efficiency; in this connection Dugald Clerk states that thermal efficiency is halved if combustion is in progress during the whole expansion stroke. This explains why an ordinary gasolene motor, built to run at fairly high speed, makes a very unsatisfactory showing when alcohol is used for fuel. Repeated tests under practical conditions have shown that though a gasolene motor can readily be made to run on alcohol, once it has been started and warmed up on gasolene, it will consume a great deal more fuel per horse-power delivered, and at fairly high speeds there will be a falling off in efficiency because the piston speed is too high for the rate of combustion of the gas; the charges may even continue to burn after leaving the cylinder through the exhaust valve. The fact that alcohol is in use in slow-running, high-compression stationary motors showing high efficiency—running at not more than 350 revolutions per minute, and often less—is merely further evidence of the futility of attempting to obtain good results by using a slow-burning fuel in a fast-running motor. But the price of gasolene is increasing and the source of supply is disproportionate to present, to say nothing of future, demands; while only a brisk demand is required to bring about the production of denatured alcohol at a price per gallon below that of gasolene. The problem is, therefore, to devise means for using alcohol in existing gasolene motors without radical change. This problem has been attacked in an ingenious way by Thomas L. White, his solution of it consisting in mixing the alcohol vapor and air with acetylene gas and burning this mixture in the cylinder.

Acetylene gas, as is well known, detonates on ignition, with the force and suddenness of a high explosive, being the very antithesis of alcohol in this respect, and a judicious proportioning serves to raise the combustion rate to a degree that causes the new mixture, called by the inventor "alkoethine," to produce results equal and even slightly superior to gasolene, under gasolene conditions. The apparatus used consists of an ordinary carbureter, such as is used for vaporizing gasolene, and a chamber for holding calcium carbide; it is illustrated more or less diagrammatically by Fig. 1. The carbide chamber is a plain vertical cylinder, with a wire netting about a third of its hight from the bottom to carry a bed of carbide. A pipe from the carbureter throws the mixture of alcohol vapor and air down on the carbide. Commercial alcohol contains, usually, about 10 per cent. of water; this is instantly absorbed by the carbide, and acetylene gas is given off. The resulting mixture passes directly to the cylinder by way of the central pipe, opening below the carbide bed. Thus the alcohol is dehydrated and the liberation by combustion of acids injurious to the valves and cylinder walls is prevented, while the heat resulting from the decomposition of the carbide has the effect of completely vaporizing the alcohol and causing an intermolecular union of the constituents of the fuel gas. Alcohol has no affinity for calcium carbide and passes unaffected; the water alone is held out. All air used is heated by the exhaust pipe. At the side of the carbide chamber is a relief vent to prevent damage by back-firing.

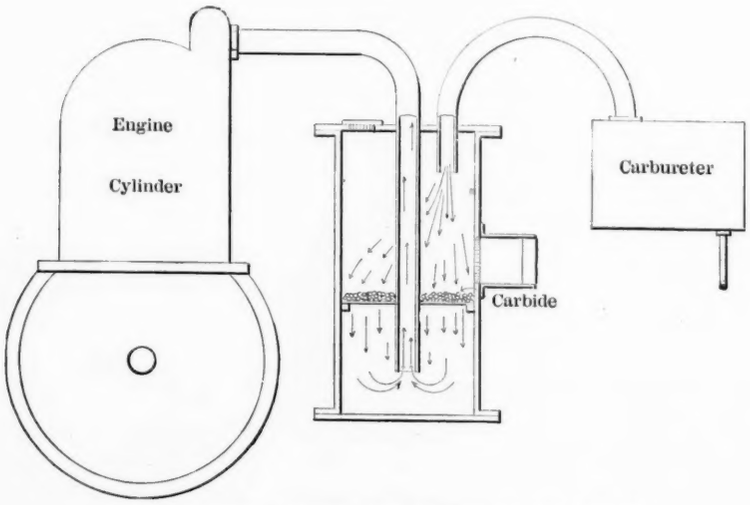


FIG. 1. USING ALCOHOL AND CARBIDE IN GAS ENGINE

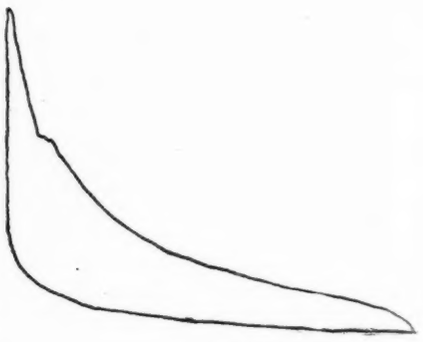


FIG. 2

As the percentage of water usually found in alcohol is insufficient to generate the proportion of acetylene required to cause the desired rapidity of combustion in the cylinder, the expedient is adopted of adding more water. A total of 18 per cent., by volume, of water is required to generate sufficient acetylene gas to produce results that are practically identical with those obtained from gasolene. More water produces still more rapid combustion and when the proportion is about 25 per cent., the resulting gas explodes in the cylinder with the peculiar ringing sound that is heard when pure acetylene and air are used in this way. This marks the danger point. The most economical proportion is approximately 18 per cent. of water. The bulk of the fuel and its efficiency under the conditions existing is increased at the expense of carbide, but the results obtained outweigh the extra cost. Carbide, in 100-pound lots, costs a trifle more than 3 cents a pound. Denatured alcohol at present must be bought at drug-store prices, varying from 35 cents to 45 cents a gallon; but with alcohol at the same price per gallon as gasolene and carbide at its present market price, a motor can be run on "alkoethine" at the same cost per horse-power as on gasolene.

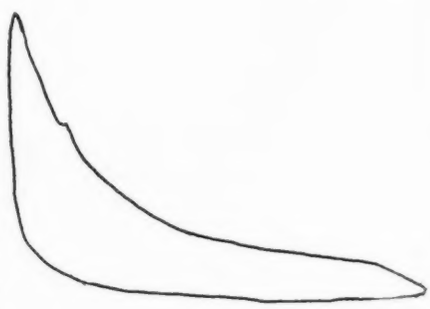


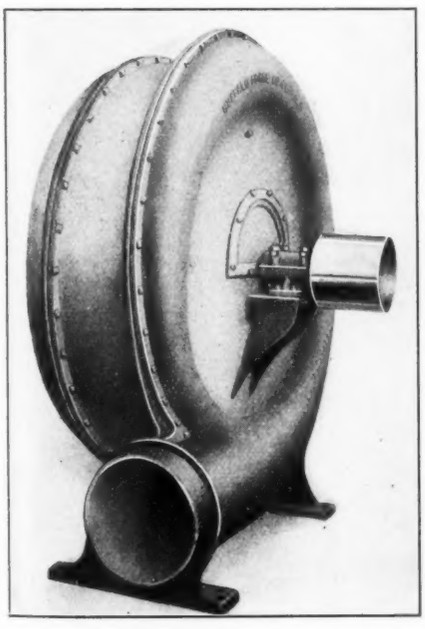
FIG. 3

This system was tested recently by Joseph Tracy, on a small automobile en-

gine direct-connected to a dynamometer, and the diagram shown in Fig. 2 was obtained. It will be noticed, upon comparing this diagram with Fig. 3, that the new fuel gives results very similar to those obtained with gasolene. The slightly higher rate of combustion of "alkoethine" makes it possible to obtain equal efficiency at higher speeds than with gasolene. The inventor puts forward the theory that a wave of compression spreads through the charge of gas upon the ignition of the molecules at the sparking points, and that this compression causes the ignition of the molecules of acetylene which, in turn, ignite the molecules of alcohol around them, ignition thus taking place at myriad points simultaneously. Tests seem to indicate the correctness of the hypothesis.

Centrifugal Two-stage Blower

The ingenuity of designers of centrifugal blowers has always been directed to attaining high pressures because of the simplicity and quiet running of this type of compression as compared with positive or displacement types. A practical limit to speed having been reached, the next course was to select a diameter sufficient to provide the peripheral velocity necessary to give a "velocity head" of the required static pressure. Instances have been known of blast wheels of steel pressure blowers having been successfully made 110 inches in diameter, and only three inches in width at the periphery, but this was necessitated by high pressure demanded when the machine was direct-connected to a low-speed motor.



CENTRIFUGAL TWO-STAGE BLOWER

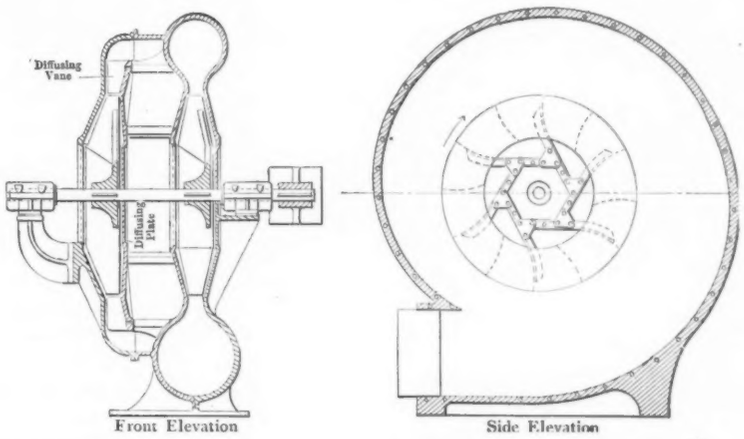
The two-stage centrifugal blower described herewith was designed to give sixteen ounces pressure when belt driven by a 50-horse-power motor at a speed of 1420 revolutions per minute. This speed, it is stated, corresponds to that which would produce only eight ounces pressure if a single-stage blower was used; in other words the two-stage scheme halves the head handled by each blast wheel with the corresponding reduction in speed required. In the design of blast wheels as made by the Buffalo Forge Company the formula used is:

$$P = \left(\frac{V}{5200}\right)^2$$

where P = pressure in ounces per square inch and V = the velocity of the tips of the fan blades in feet per minute. The machine shown in the illustrations was designed not to give an excessively high pressure (for sixteen ounces have been obtained from single-stage machines), but to give a good pressure at a low speed, with the attendant long life and operating efficiency.

The two blast wheels are hung on a 2½-inch shaft running in self-alining, ring-oiling bearings. The hubs of the wheels are hexagonal in shape, of cast-iron and each firmly held in place on the shaft by a key and two set-screws. The spokes, which are of 1½ × 1½ × ¼-inch angle-iron, six in number, are riveted to the sides of the hexagonal center and to each other by triangular lap pieces, affording a very rigid spider construction. These blades are curved forward to produce a higher pressure for a given speed, and they are made tapering from the periphery to permit a constant acceleration, without the formation of eddies in the passing air. The side sheets of the blast wheel are flared at the inlet diameter for rigidity and as this projection is followed closely by the casing section the smallest possible space for leakage is allowed.

The air discharged from the blade tips of the first wheel on its way to the inlet of the second wheel passes through "diffusing vanes," which are placed at an angle of about 115 degrees with the radius. These vanes are designed to prevent loss by eddy currents and shock, and to enable the kinetic energy contained in the air, due to its velocity, to be transformed into potential energy or pressure. The diffusing vanes consist of 20 plates of No. 14 sheet-steel, which are cast into a retaining plate called the "diffusing plate,"



SECTIONAL VIEWS OF BUFFALO FORGE COMPANY'S CENTRIFUGAL TWO-STAGE BLOWER

composing the interior side of the casing of the first stage. After passing through the second blast wheel the air enters a housing, having a cross-section which increases in an arithmetical ratio up to the full area of 17¾ inches diameter of the discharge.

A peculiar difference exists between the two-stage blower and a two-stage centrifugal pump, because of the difference in density of the fluid handled. If a pump were constructed on similar lines with both inlets exhausting along the shaft in the same direction, a good-sized thrust bearing would be needed to withstand the tendency of the shaft to move parallel to its axis. In the blower, however, a method of solution is possible which cannot be resorted to in pump design, if maximum inlet efficiency is to be obtained. This thrust tendency is counteracted by cutting away (on each blast wheel) a portion of the side plate of the wheel which would otherwise inclose a chamber between the side of the casing away from the back of the inlet and the back of the impeller. The portion cut away is of a diameter approximately that of the inlet and serves to release the pressure which would otherwise accumulate by leakage through the clearance space between the wheel and the casing.

The machine shown in section measures over all 4 feet 7 inches x 6 feet 3 inches x 6 feet 10 inches, and will furnish blast to melt all the iron a 64-inch cupola is capable of handling. This blower is under construction at the shops of the Buffalo Forge Company, Buffalo, N. Y., for the Fanner Manufacturing Company, of Cleveland, Ohio.

A Large Rolling-mill Engine

One of the most interesting features of the new continuous structural mill recently placed in operation at their South side works by the Jones & Laughlin Steel Company is the engine equipment. The three engines that operate this mill were designed and built by the C. & G. Cooper Company, Mt. Vernon, Ohio.

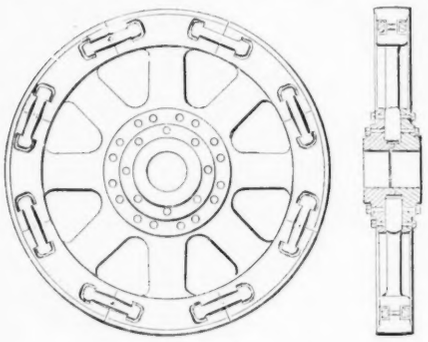


FIG. 3. THE FLY-WHEEL

They represent the most modern practice in engine design for this class of work, and are of the tandem compound-condensing Corliss type. The short time in which these engines were built was very remarkable. Two of the engines were shipped in seventy days after receipt of order, and the third, a 5000-horse-power machine, in less than ninety working days. The last named engine was an odd size, and had to be especially designed, requiring new drawings and patterns throughout. All the engines are directly connected to roll trains by means of steel couplings, and are equipped with variable-speed governors so arranged that the speed may be varied between 60 and 110 revolutions per minute while in operation. The normal speed, however, is 80 to 90 revolutions per minute.

Fig. 1 shows the 43 and 74x54 roughing-mill engine in the course of erection in the shops of the C. & G. Cooper Company, and Fig. 2 shows the same engine in more complete detail. The main shaft of this machine is of Bethlehem hollow hydraulically forged steel, 27 inches diameter in the wheel fit, and 25 inches diameter 50 inches long in the main journals. The fly-wheel is 18 feet diameter, weighs 200,000 pounds and is of the built-up or segmental type, as shown by Fig. 3.

The outboard pillow block is of the massive pedestal type, as shown in Fig. 4. This pillow block is held down by twelve 3-inch foundation bolts, and further locked into the foundation by means of a transverse rib extending across the bottom of the block. A heavy beaded edge and oil channel is cast solid with the block on all four sides for the purpose of catching oil from the journal, and also from the bronze thrust bearing between the coupling and the pillow block. This journal is the same size and design as that of the main bed-plate, being provided with removable shells and adjusting wedges, which have full-length bearings against both side gibs.

As will be seen from Figs. 1 and 2, the guide barrel has a full-length support cast solid with it. This guide barrel is of extra heavy cross-section, and is bored in cylindrical form. The bed is of the Tangye rolling-mill type, with extra depth underneath the main journals to insure maximum strength and rigidity. A heavy base plate extends under the engine and unites with the bed-plate in a bolted joint. This base plate is 15 inches deep, is strengthened by heavy longitudinal and transverse ribs, and has a continuous beaded edge and oil channel. The cylinders and guide barrel are connected to the base plate in planed joints, ample provision being made for relieving the engine of any undue strains of expansion and contraction.

The low-pressure piston floats on a hollow piston-rod supported between the

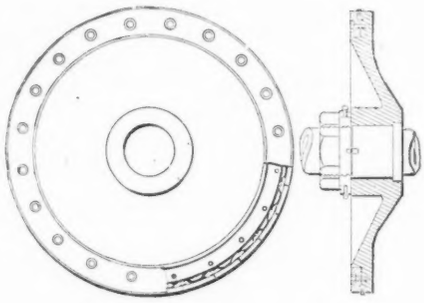


FIG. 4. LOW-PRESSURE PISTON

main and intermediate crossheads, and is the steel umbrella type provided with follower and self-acting steam packing ring. This piston does not come in contact with the cylinder except at the packing ring. The construction of this piston will be more clearly understood from Fig. 4. An intermediate crosshead and guide barrel is provided between the high- and low-pressure cylinders as a support for this floating piston. This crosshead also forms an accurate connection between the high- and low-pressure piston-rods. The piston of the high-pressure cylinder is of the center-ring type, with means for central adjustment, and has follower, self-acting packing ring and bull

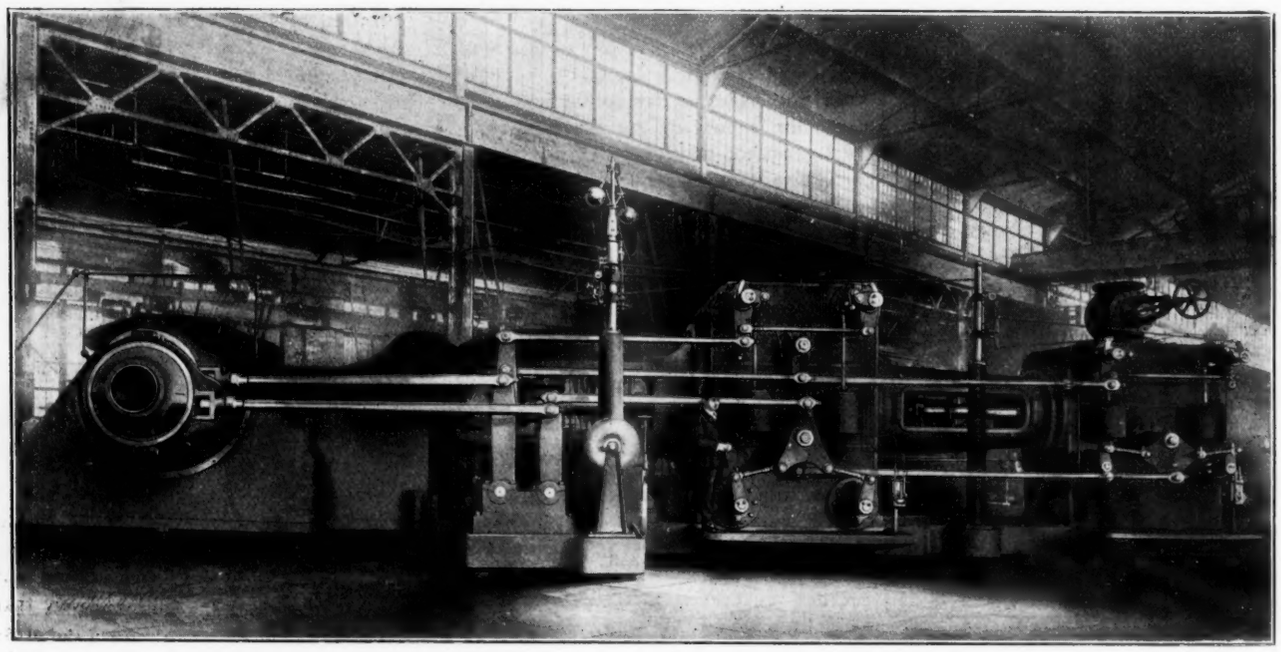


FIG. I. A LARGE ROLLING-MILL ENGINE

rings which cover the entire face of the piston. These bull rings are fitted with alternate sections of composition metal and grooved for carrying the oil, as shown by Fig. 5.

The valves are of the multi-ported type, ground on dead centers, and polished on all wearing surfaces, which insures a steam-tight fit. The steam valves of the high-pressure cylinder, and admission valves of the low-pressure, are direct-driven from the rocker arms, and are under control of the governor to 75 per cent. of the stroke. The vacuum pots are of the Cooper improved piston oil-sealed type. The valve gear is of steel, having all bearing surfaces lined with phosphor bronze or council metal.

The finishing rolls are driven by a 38 and 66x48 engine, having main shaft 25 inches in diameter in the wheel fit, and 23 inches diameter and 46 inches long in the main journals. The fly-wheel is 18 feet diameter and weighs 160,000 pounds. This engine throughout is similar in construction to the roughing-mill engine.

The two straightening machines are driven by a 22 and 40x42 engine, which is also similar in construction to the larger engine. The fly-wheel on this machine is 16 feet diameter, and weighs 60,000 pounds.

The three engines are equipped with a gravity automatic oiling system also furnished and installed by the C. & G. Cooper Company. The oiling systems for the roughing and finishing engines are each of ample capacity to serve both engines, and are so piped up that should the filter or pump of either system be out of commission for cleaning or repairs, the other system may serve both engines. The cups and fittings are of the needle-valve type, so constructed that the glass reservoir of the oil cup is not under pressure, but the oil supply and pressure are against the needle valve. Each cup is independent in its operation, so that in case one cup is out of order it will in no way affect the other cups of the system. The cups and fittings are so arranged that should the entire system be temporarily out of service, the cups will operate as ordinary sight-feed cups by simply throwing the lever on top of the cup and fill by hand. This system insures ample lubrication at all times, with a minimum expense since all oil is caught, returned to the oil filter, and used over and over again.

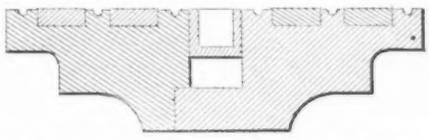


FIG. 5. LONGITUDINAL SECTION THROUGH HALF OF HIGH-PRESSURE PISTON

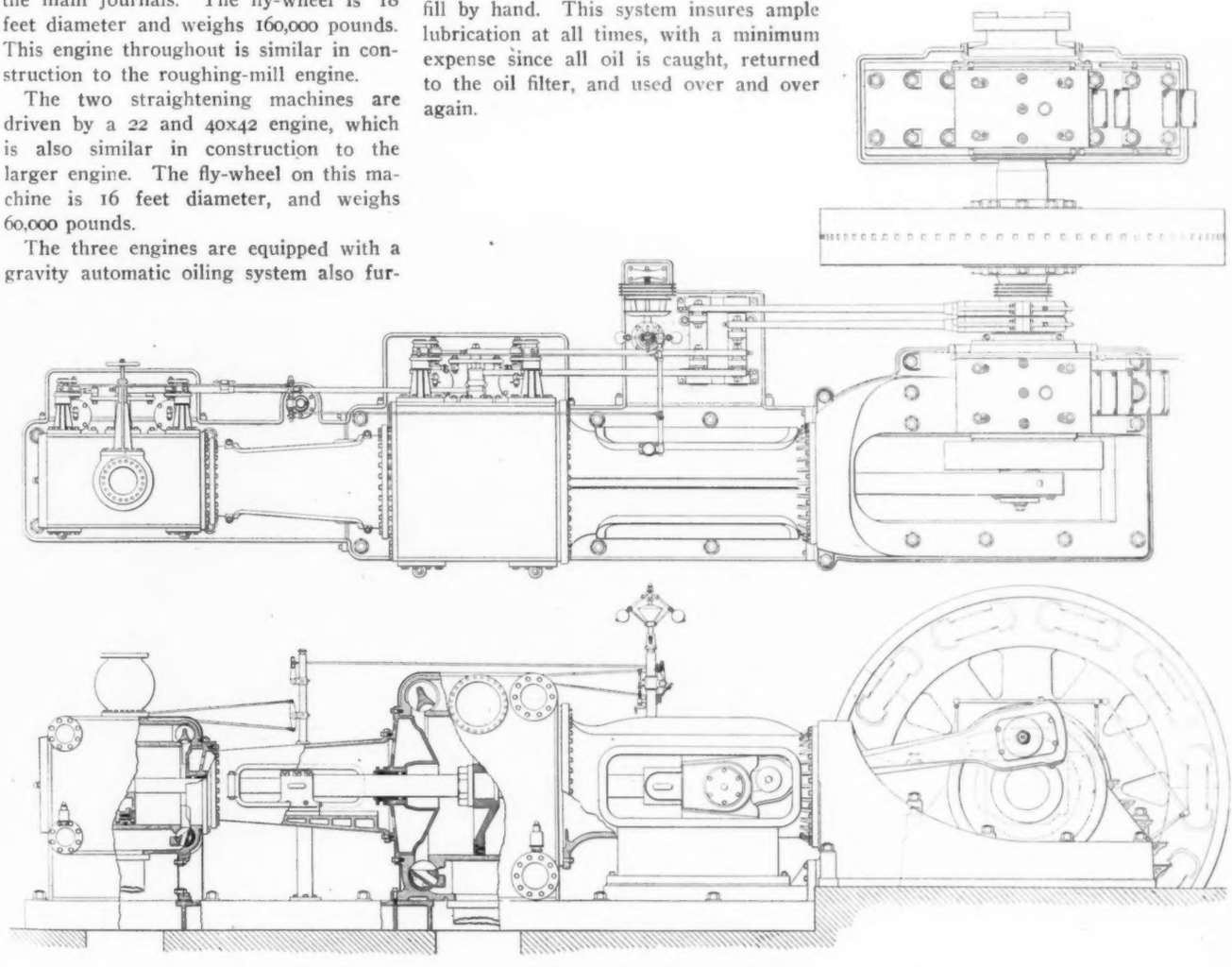


FIG. 2. PARTLY SECTIONAL ELEVATION OF COOPER ROLLING-MILL ENGINE

The engines made their trial run of nine hours up to speed the day they were started without showing any tendency to heat at any point. They have now been in successful operation about twelve months, turning out a large tonnage of bars, beams, channels, and other structural shapes.

Dust Determinator for Blast-furnace Gases

The cleaning of furnace gas is becoming of great importance in the economical manufacture of iron and steel. It not only is necessary to clean furnace gas of practically all the solid matter in order to use the gas successfully in internal-combustion engines, but it is found very desirable and productive of economy to clean the gas used in the stoves of a blast furnace. The dust and dirt from uncleaned gas amount to 12 to 30 grams per cubic meter, 5 to 13 grains per cubic foot, and while the large particles of ore, limestone and coke in the gas are precipitated by gravity into the pockets of the flues, the fine and impalpable dust will remain suspended in the gas-like smoke in atmosphere and can only be removed by washing or filtering through some medium which allows the gas to pass and retains the solid matter.

When furnace gas is used for fuel in the cylinder of an engine, a very small amount of dust is prohibitive as it, nat-

urally gritty, will unite with the oil of lubrication, forming a pasty mass which produces an abrasive effect only excelled by oil and emery. As 75 per cent. of the dust is metallic oxide, it, when subjected

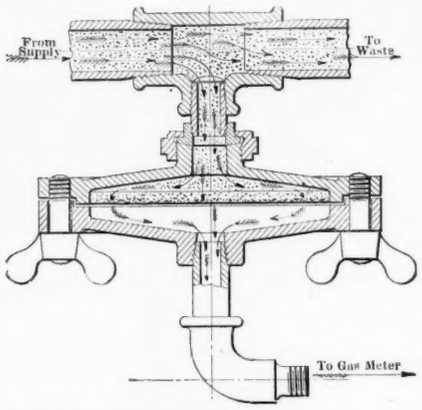


FIG. 1. SHOWING FILTER HOLDER IN SECTION

to a temperature of 3000 degrees Fahrenheit, the heat of inflammation, will be precipitated as iron or steel.

Gas used in internal-combustion engines should not have over 0.02 gram of dust per cubic meter, 0.009 grain per cubic foot, or the wearing of the engine cylinder will be excessive. When used under boilers for making steam, the freer the gas from solid matter the better. The efficiency of gas-fired boilers depends as much on the side of the tube next to the fire being clean as the side surrounded by the water.

The purpose of the apparatus under review is to determine the amount of dust carried by the gas. Such an instrument must be simple, its accuracy unquestionable and its design such that the determinations may be made hourly or as often as desired. The ordinary method of determining the dust in air or gas is to make a filter of a glass tube filled with absorbent cotton through which the air to be filtered flows. The gas is measured through a test meter and the cotton is weighed before and after. This method might give accurate results if the cotton always had the same density throughout the tube and were not hydroscopic. The cotton may be packed in so loosely that some of the dust will work through and unless the cotton is carefully dried over calcium chloride and weighed several times, a long and tedious process, errors will naturally arise. Experiments have shown that if two cotton-filled tubes are used in tandem, the second will increase in weight, showing that some of the impalpable dust is not retained by the first tube.

The filtering medium for the apparatus herein described is simply a diaphragm of white filter paper through which the gas percolates, but on account of the minute interstices of the medium, every atom of dust or dirt remains behind. The side through which the gas enters becomes the color of the dust, while the other side remains uncolored. When two filters are used in tandem, the second does not increase in weight, showing that no dust permeates such a filtering medium. The velocity of gas through the filters and a test meter which has but a one-quarter-inch pipe is not rapid and if the instrument is erected some distance from the main supply pipe, the deposition of dust on the way to the filter will cause an appreciable error.

In the apparatus described, a three-quarter-inch pipe passing across the top of filter holder, Fig. 1, allows a considerable quantity of gas, at a fair velocity which keeps the dust stirred up, to pass the opening to the filter so that the amount filtered out must be indicative of the total dust in the gas. By keeping continuous records of the dust in the gas before and after cleaning, the efficiency of the cleaners can be maintained. A check on the operation of the furnace is possible and the minimum wear of the engine cylinders is insured. A record of the condition of the furnace gas is essential in its commercial use.

Fig. 2 shows the complete self-contained determinator, which consists of a portable case containing an accurate test meter, two filter holders (in section in Fig. 1), complete cross-connected three-quarter-inch brass piping, so that the gas to be tested may flow over the mouth of either filter, and hose connections allowing the gas passing through the filter paper to be accurately measured through the test meter. When the percentage of moisture in the gas is desired, the gas is passed through a cooling coil where most of the moisture is condensed and precipitated in a collecting bottle. After passing the cooling coil the gas is passed through three or four bottles of calcium chloride, removing effectually any further moisture in the gas before it is metered or its calorific value is determined. When the determinations are merely for finding the percentage of dust, the cleaned gas, after leaving the meter, is mingled with the main supply and burned or wasted to the atmosphere. The cleaned, dried gas may be passed through an automatic calorimeter by which the B.t.u. are determined as well as the hydrogen in the gas. A complete record of the dust and calorific value is an indication of the internal furnace conditions, desirable in the economical manufacture of steel.

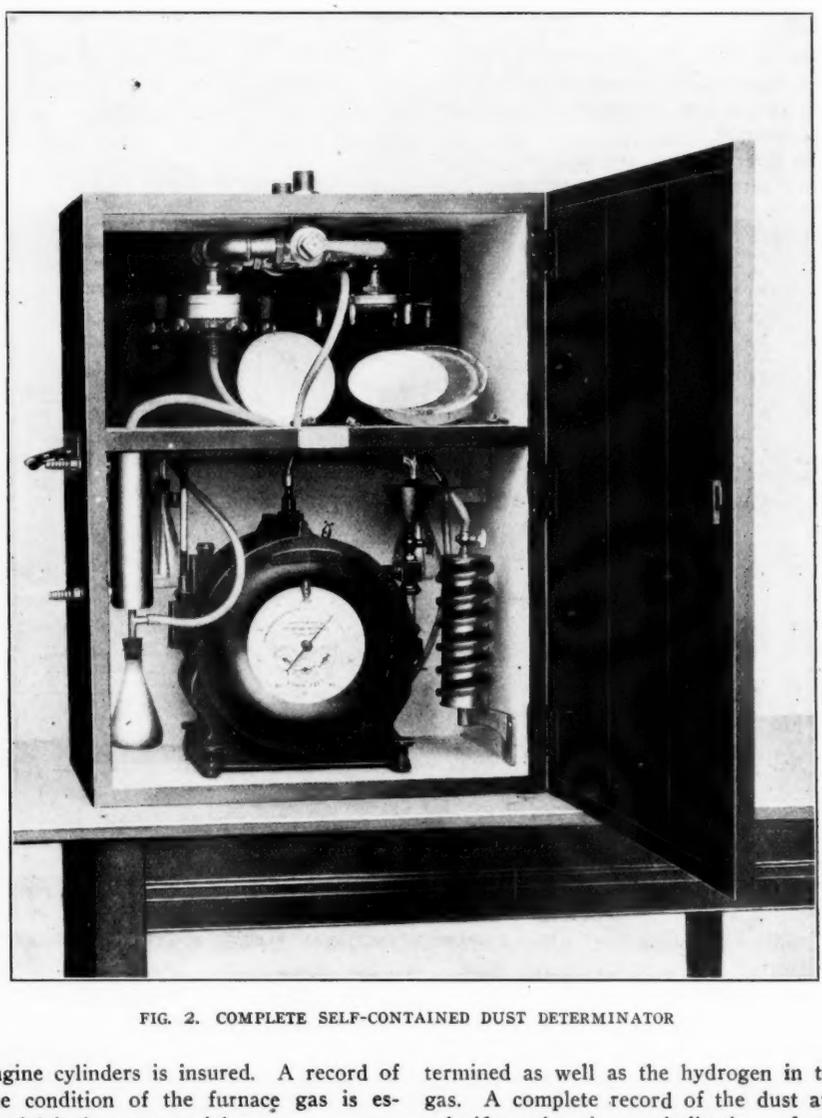


FIG. 2. COMPLETE SELF-CONTAINED DUST DETERMINATOR

By using two filter holders continuous determinations can be made. By the proper manipulation of the valves gas can be passed through either filter while the dust collected in the other per cubic foot of gas burned is being ascertained. On account of the moisture in the gas softening up the filter paper, a wire gauze is inserted under the filtering medium, which prevents the weight of the dust from tearing it. As the deposited dust and filter paper remain more porous if kept warm and dry, an incandescent lamp or candle is burned under the filter holder in use. The inlet and outlet pipe for gas passes through the case which is provided with a door and lock and may be left running for twenty-four hours if desired, though hourly readings may be obtained if the variations of the dust under different conditions are desired. The proportion of gas wasting and going through the meter is adjustable and can be regulated to suit the conditions and location of the apparatus relative to the flue from which the sample is taken. If the burning or wasting of the gas flowing by the filter mouth is not desirable, it may be piped back into the gas flue in such a way as to maintain a circulation through the shunt.

The operation of the determinator is as follows: Locate the instrument as close to the flue containing the gas as practicable. Run a three-quarter pipe from the flue to the inlet pipe at the case; run the waste pipe where desired. Level and fill the test meter with water until it rises to the level indicated in the glass by the pointer. Place the filter paper in the holder and tighten the thumb screws (this may be done in the laboratory where the filter paper is weighed if desired and the holder can be connected up by union). Note the reading of the gas meter, the date and hour; write the same on the filter holder being used. After a certain time, depending on the amount of dust in the gas, the meter is read and the gas by-passed through the other filter which had been previously prepared. Remove the filter holder and determine the amount of dust collected as follows: The filter paper having been previously dried and weighed should be carefully dried again by subjecting it and the dust attached to the same temperature—not less than 212 degrees Fahrenheit. This will drive off the moisture and the difference between the weight of the clean paper and the dust-covered paper will give the net amount of dust, which divided by the cubic feet of gas will give the grains or grams per cubic foot.

To get the percentage of moisture, weigh the water precipitated in the inverted flask below the condenser and by weighing each flask of calcium chloride (the weight of each having been noted before the test began), the percentage of moisture is readily obtained. In order to be sure that all moisture has been extracted, the last flask through which the gas passes should not increase in weight.

A pressure gage and thermometer at the meter will allow of a reduction of the meter gas to standard gas, should this be desired for comparison. As 15 to 25 per cent. of the dust in furnace gas is coke, it is not advisable to determine the total dust by incineration, though this method is used where the solid matter collected contains no combustible. When air under atmospheric pressure is analyzed for foreign matter, a water ejector or a rotary hydraulic-pressure transformer is used to draw the air through the filter and meter.

The dust determinator is manufactured by the Sargent Steam Meter Company, First National Bank building, Chicago.

Wile Bituminous Gas Producer

The Wile Power Gas Company, Rochester, N. Y., has brought out a power-gas producer designed especially to burn bituminous coal and capable of burning any other fuels without any change in the apparatus.

Fig. 1 is a sectional elevation of the generator and shows that it is of the double-zone class. In the upper zone the fuel is burned to coke and air gas is distilled from it; the coke passes to the lower zone, where richer gas is distilled by the introduction of steam with the air blast, as in the ordinary anthracite producer.

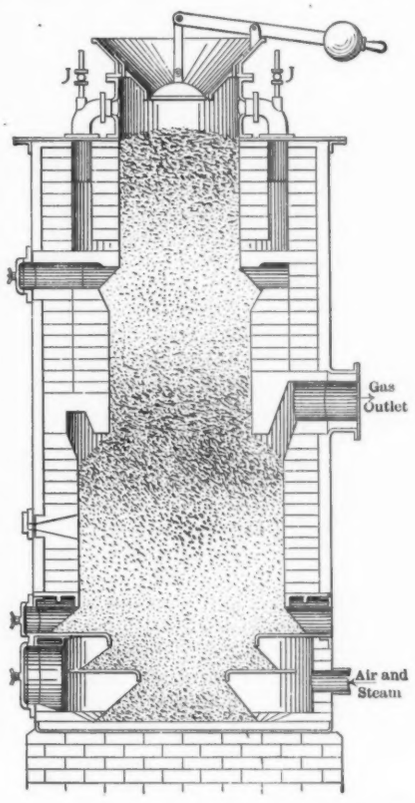


FIG. 1. SECTIONAL ELEVATION OF GENERATOR

At *J* and *J* are located blower jets which cause a circulation of the volatile gases, which rise from the green fuel. These are blown down into the incandescent part of the fuel bed and pass up through the bed again until they become "fixed." The fixed gases distilled from the coke in the lower zone pass out through the outlet below the upper zone As the gases accumulate in the upper

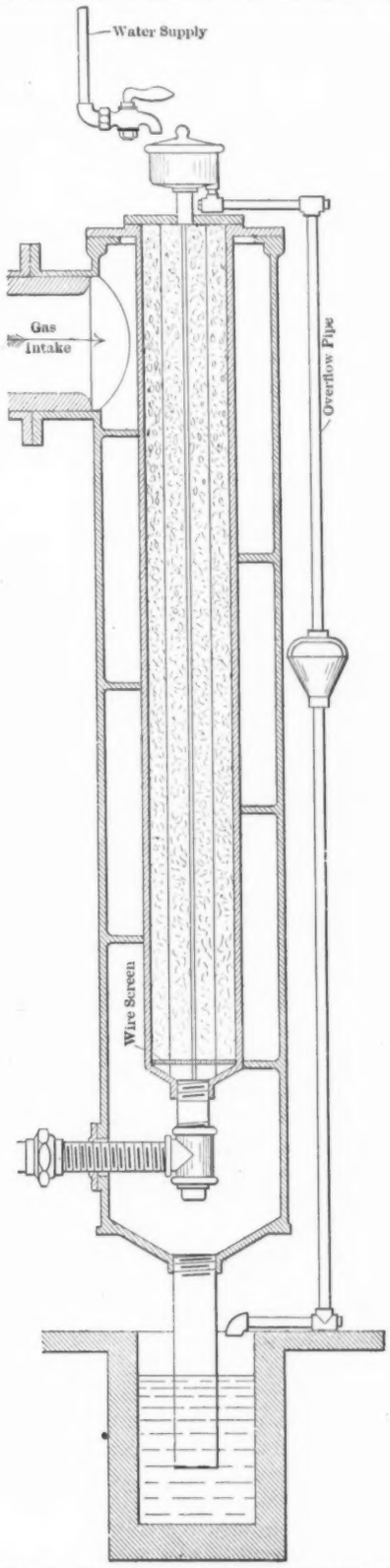


FIG. 2. SECTIONAL ELEVATION OF VAPORIZER

zone, the pressure there increases until some of the gases are forced down toward the outlet and drawn out by the suction from the engine. The carbon dioxide drawn from the upper zone to the outlet picks up carbon in its passage through the hot coke and becomes carbon monoxide.

The temperature of the incandescent fuel in the upper zone is much higher than that in the lower zone because no steam is introduced. This higher temperature is relied on to burn out the tar and fix the volatile gases, and the circulation facilitates these processes.

The generator is provided with poke holes in the sides, as usual.

The admission of steam with the air fed under the lower zone is controlled automatically by the suction of the engine. Fig. 2 is a sectional elevation of the vaporizer. Water is delivered to the vaporizer from a small constant-level reservoir, in which the water level is just below the level of the delivery outlets. This construction is shown in Fig. 3. The direct-supply chamber *A* is water-sealed from the surrounding reservoir chamber *B*, so that the engine suction produces a partial vacuum in the chamber *A*; the atmospheric pressure in the chamber *B* raises the water level in the inner chamber and thereby delivers water to the pipe leading to the vaporizer. The greater the engine suction, of course, the more water will be delivered to the vaporizer. The water level in the chamber *B* is maintained constant by supplying more water to the reservoir than is needed and providing an overflow outlet at the desired level.

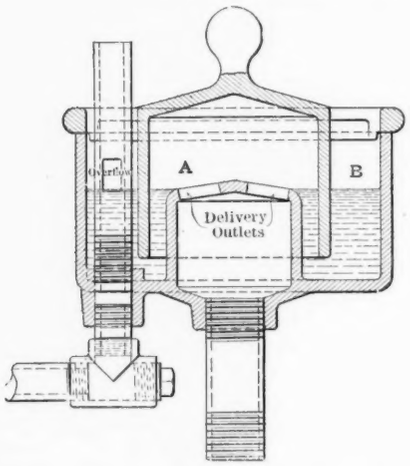


FIG. 3. SECTIONAL ELEVATION OF WATER-FEED REGULATOR

The gas, after passing through the outer shell of the vaporizer, goes through the usual wet and dry scrubbers to the engine. Fig. 4 shows the layout of the complete producer plant.

The Vulcan Soot Cleaner

The purpose of the Vulcan soot cleaner, which is illustrated herewith, is to remove soot, dust and ashes from the heating surfaces of a boiler, while the boiler is in use, instead of having to draw the fire and wait until the boiler cools before it can be cleaned. At the same time it obviates hand cleaning. The principle employed is that of blowing, at high velocity, jets of dry steam or compressed air against the soot-covered parts, the process involving on the part of the operator merely the opening and closing of a few valves. This may be done when the fire is at full heat, and with every door and opening tightly closed, thereby practically eliminating the danger of condensation, at the same time enhancing superheating conditions. The cleaner is designed in various forms to fit the needs of different types of boilers.

As ordinarily applied to a Stirling water-tube boiler, for instance (Fig. 1), the appliance consists of a stationary system of piping which branches from a 2-inch main steam line and enters the boiler setting from one side at appropriate points. Each branch, which supplies several of the entering or blower pipes, is controlled by one valve. These blower pipes are so perforated that jets of steam or air are shot in different directions between the rows of tubes, as well as against the surfaces of the headers and drums, and the tops of the arches and baffle plates. At points where the heat is greatest, and there is the least accumulation of soot, the pipes

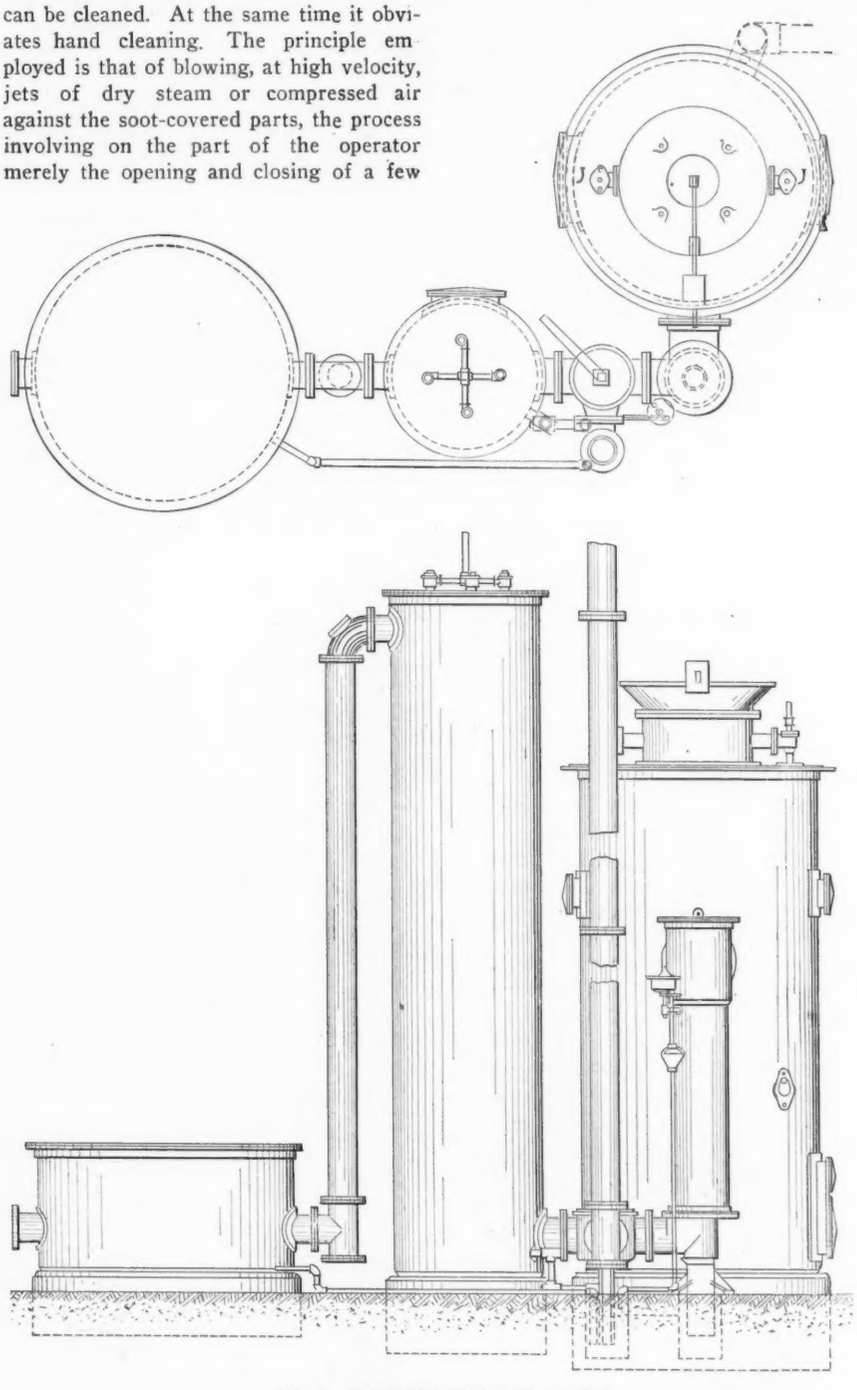


FIG. 4. COMPLETE PRODUCER PLANT

are short and therefore less exposed to the heat; otherwise, the pipes are long and calculated to reach the farthermost surfaces of the boiler. When the system is not in use all the valves are left open, except that in the main steam line, of course; this permits sufficient cold air to enter through the drain valve to keep the pipes from being burned out.

To operate the cleaner all valves are closed, except the drainage valve, and the valve to the steam line is opened. When the drainage valve shows dry steam it is closed and the branch valves are operated in turn, blowing back with the draft. When the blowing is completed the main valve is closed and the drainage valve and the valves operating the blowers are opened and left open, as stated; also an air valve, to prevent accumulation of dead air in the interior pipes. The entire operation takes from seven to ten minutes and, to get the best results, should be repeated every six or eight hours, it is stated.

It is pointed out that by using certain of the blowers in time of emergency, a powerful forced draft results. Then, too, the soot and ash which have been blown back into the combustion chamber can be removed by means of an "ejector," which may be connected with the cleaner system, and which blows out the accumulations into the ash-pile or into a retort, as desired. Or if there is no objection to a little smoke, a spray system may be used to send the soot and dirt out through the stack.

small nozzle, as illustrated in Fig. 2. These branches are formed of standard fittings, tees, and globe valves. In operating, one branch is opened at a time, working toward the receptacle, and the soot is blown up the stack.

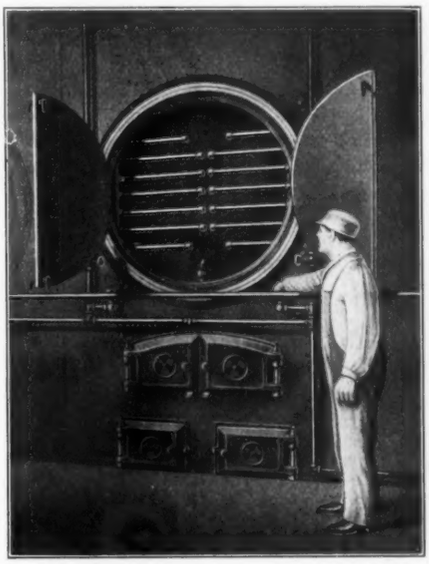


FIG. 3. FRONT-END BLOWER FOR HORIZONTAL TUBULAR BOILER

For cleaning horizontal return-tubular boilers either of two methods is employed: by blowing from the rear of the boiler toward the front, or the reverse. The principle of the rear-end blower is

arm swings across the rear of the boiler and turns within a sleeve set in the rear wall. On the outside of the wall is a long handle with swivel coupling for the purpose of turning the cleaner. The two valves for controlling and draining are also on the outside, so that it is not necessary to open the smoke doors to operate the blower.

To operate, first open the drain valve, then close it, open the supply valve and give the handle a few turns. The operation requires about three minutes. The inner arm or pipe is protected by thick asbestos covering, an air space and a wrought-iron sleeve.

The front-end blower, Fig. 3, consists of a series of horizontal pipes, one across the end of each row of tubes, with a nozzle opposite the center of every tube. These pipes are supported by and supplied with steam from two swinging manifolds contained within the smoke-box, one on either side. When not in use the pipes may be swung away from the tubes by the mere pulling of a handle. Each manifold is divided into two parts, each part controlled by a valve on the outside, so that in blowing the boilers one-fourth of the tubes are cleaned at a time, and in this way the draft is not obstructed. The piping on the outside is so arranged that the whole system is drained thoroughly.

To operate the front-end blower the operator shoves a handle, operates the valves, and the steam pressure from the boiler does the rest. After the four operating valves have been blown for about a minute each, they are again blown in succession for a few seconds to blow back any fine matter that may have been carried in by the draft. The "ejector" is also applicable to this type of boiler for cleaning the combustion chamber.

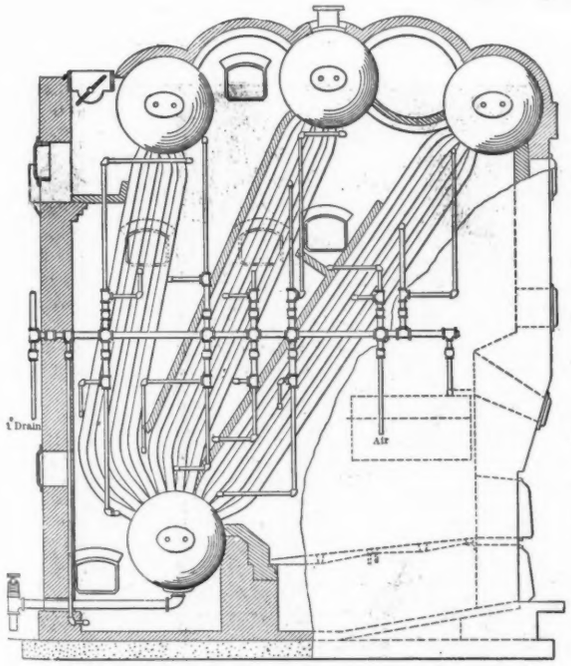


FIG. 1. VULCAN SOOT CLEANER APPLIED TO STIRLING WATER-TUBE BOILER

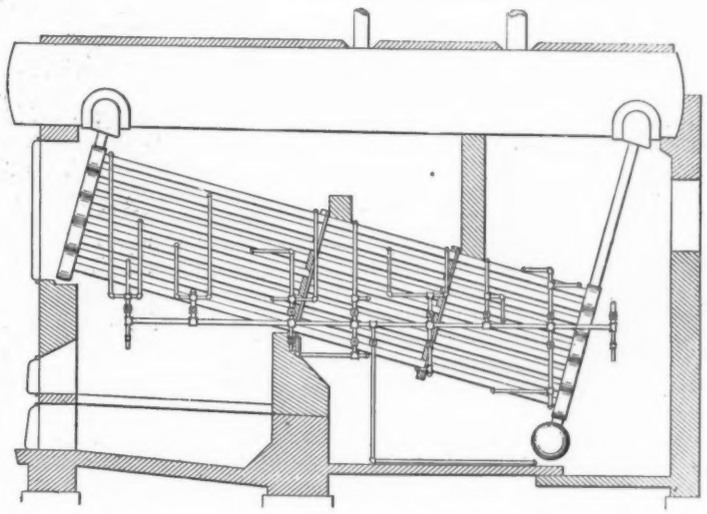


FIG. 2. VULCAN SOOT CLEANER APPLIED TO BABCOCK & WILCOX BOILER

As installed on the Babcock & Wilcox boiler, the Vulcan cleaner consists of a horizontal pipe taking steam from the steam drum of the boiler, having branches to the regular dusting doors ending in a

to blow with the draft and to throw a direct jet of dry steam or compressed air into every tube. This is done by a movable arm, or steam pipe, which contains a number of nozzles for blowing. The

The Vulcan soot cleaner is manufactured by the Vulcan Soot Cleaner Company, Du Bois, Penn.; it is the invention of Wm. Eichelberger, superintendent of the company's works.

Power Plant Machinery and Appliances

Original Descriptions of Power Devices
No Manufacturers' Cuts or Write-ups Used

MUST BE NEW OR INTERESTING

"Argand" Arches and Jambs

The "Argand hot-air diffusion arches and jambs" herewith illustrated are designed for a twofold purpose: to minimize the liability of the burning out of arches and jambs contiguous to the feeding-door of the furnace, and to prevent smoke as far as possible, thereby enhancing fuel economy, by employing an old principle in a new guise. Regarding the life of the parts around the furnace door, the trouble of having to rebuild them frequently, not to speak of the expense, has impressed itself very vividly upon all interested. It was while seeking a remedy for these evils that the inventor of the "Argand" devices was led to the subject of better combustion, with the result that the invention is based on the assumption that ordinarily perfect combustion is not attained owing to the loss of heat units in the passage of the gases from the face of the fire-bars to the surface of the coal, due to the lack of oxygen in sufficient proprtion to insure complete combustion.

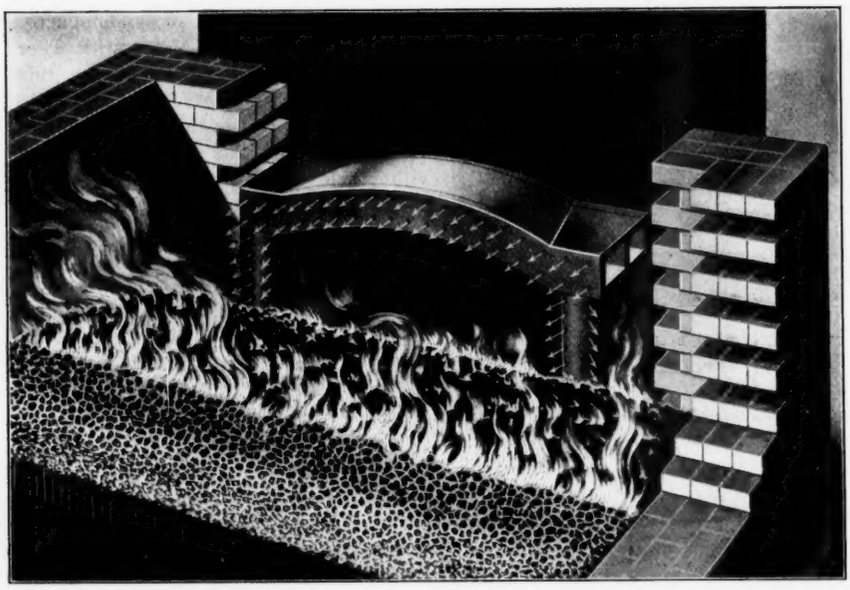


FIG. 1. SHOWING HOW AIR IS EJECTED ACROSS THE FIRE

To introduce the needed oxygen in a manner best calculated to secure this result, the designer, John Sherman Roake, conceived the idea of introducing air in jets through perforations in the back of the arch and jambs (Fig. 1). A central partition divides the arch laterally into halves, each half being sub-divided into two equal channels by a lengthwise partition which does not extend quite to the dividing wall, thus permitting the free circulation of air, which enters the front channel at either end through a flanged opening into the jamb. The air flows around the end of the partition into the rear channel and out through the perforations over the fire-bed, being expelled in jets, as indicated. Each jamb is hollow and is held to the supporting front frame by a single bolt. As a rule, draft holes for the admission of air to the jambs are provided in the front frame, just below the jamb bolts, there being one hole in each side of the frame (see Fig. 2). These bolts have swing covers which can be set to admit much or little air, as desired.

In furnaces where forced draft is used, however, the draft holes are not made in the boiler front, but are located in the dead plate, one under each jamb, as shown in Fig. 2. In this case the jambs and arch receive the same air pressure as that in the ash-pit. There is a row of perforations in the back of each jamb, through which the air enters the furnace in jets. One of the features of this combination, it is pointed out, is in the air vents being in such position as to avoid danger of becoming filled with ashes or slag. Also—and this is the main idea—the air in its passage through the jambs and arches abstracts the heat from those parts, so that they do not become unduly heated and burned out. The arch is 12 inches wide and 4½ inches deep, and both itself and the jambs are of high-grade ¾-inch cast-iron. The builders of the "Argand" arches and jambs are Wilson & Roake, corner of Front and Dover streets, New York City.

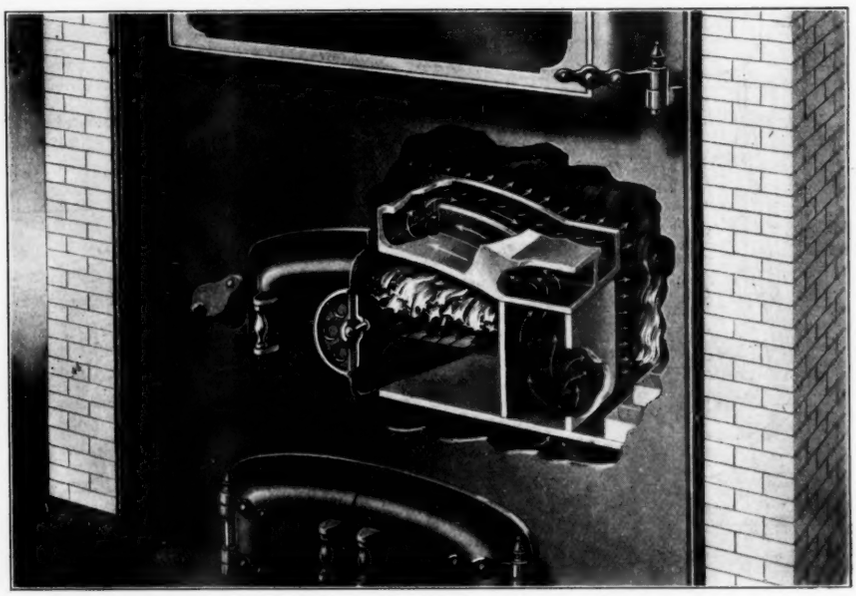


FIG. 2. SHOWING CIRCULATION THROUGH JAMB AND ARCH

"Perfection" Shaking and Chopping Grate

The accompanying illustrations show the essential features of the "Perfection" shaking and chopping grate. Its construction differs from other standard grates chiefly as to the grate surface, which is formed by the assemblage of teeth having bifurcated shanks set loosely over the bridges of the rocker bars. This gives a substantially flat grate surface, while it avoids straight slots either lengthwise or across the top, all the lines being cut up without sacrificing air space; and the air space does not change in shaking.

In assembling the teeth the short and long faces are alternated, as shown in Fig. 1, giving the grate surface a dovetailed appearance, as in Fig. 2. The teeth measure 1½ inches from crown to rocker bar, this providing a substantial "heading-course." The rocker bars are made in short horizontal sections, to minimize danger of warping; for the same reason the bearing caps are very heavy, and the center bar between the grate halves is also exceptionally strong.

Fig. 1 shows a rocker bar with teeth in position; also four loose teeth, a bearing cap and a shaker bar. The simple manner of connecting up the shaker and rocker bars will be apparent by reference to the cut, which also suggests the method of operation with lever and rocker arm. The grate is put together without the use of screws, bolts, keys, cotter-pins or fittings of any sort, other than those shown, the working parts supporting each other.

When the grate is not in motion it is locked in the flat position and the direction, as well as the formation of the crowns of the teeth, render them unlikely to be damaged by firing implements in the hands of the firemen.

This grate is made by the Perfection Grate Company, of Springfield, Mass.

The "Thermit" Welding Process

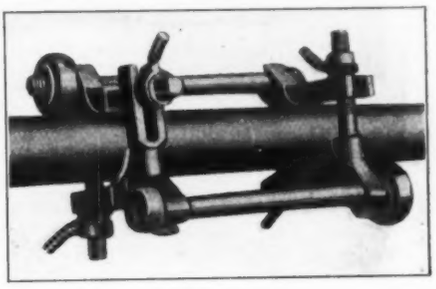
By employing the "Thermit" process tubes and rods can be butt-welded on the spot without having to resort to the use of machinery, electric power, furnace or hammering. It is especially adapted for steam, compressed air, ammonia or hydraulic piping, whether horizontal or vertical, and it is pointed out that the welded joints are as strong as the rest of the pipe.

WELDING A 1¼-INCH PIPE

"Thermit" is a mixture of finely divided aluminum and iron oxide which when ignited in one spot communicates the combustion throughout its mass without the introduction of heat or power from outside, and produces during this combustion (in about fifteen seconds) a temperature of approximately 5400 degrees Fahrenheit. During the action which generates this heat the aluminum and the oxygen contained in the iron oxide combine to form a highly superheated liquid slag, or aluminum oxide. The iron which is set free at the same time and at the same temperature sinks to the bottom of the containing vessel. The slag has three times the heat volume of the liquid iron.

The ends of the pieces to be welded are first filed, and then butted carefully together between clamps, as shown in one of the accompanying illustrations. Around the joint is placed a cast-iron mold in two plates, the upper half having a narrow gate through which the molten mass is to be poured. The "Thermit," which is contained in a flat-bottomed crucible held in a pair of tongs, is then ignited.

The highly heated molten mass is poured into the opening of the mold and heats the parts to be welded to the necessary temperature to soften the metal for manipulation.



CLAMP IN POSITION FOR WELDING PIPE JOINT

In pouring, the slag, which naturally flows first, will adhere in a thin layer to the walls of the mold, and the surfaces of the pieces to be welded, and will protect them from contact with the liquid steel which runs in last. On chilling, this layer of slag is easily knocked off. After waiting for the ends of the pipe to soften, the clamps are drawn up and the weld effected by squeezing together the plastic ends of the parts. After removing the mold and clamps, the weld is found to be

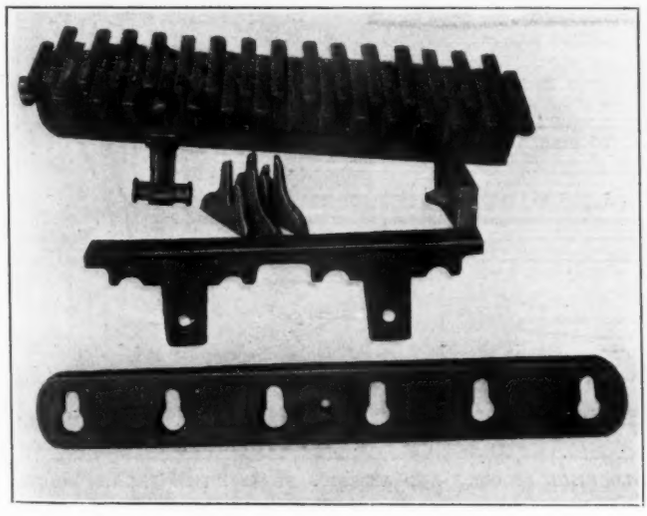


FIG. 1. COMPONENT PARTS OF PERFECTION GRATE

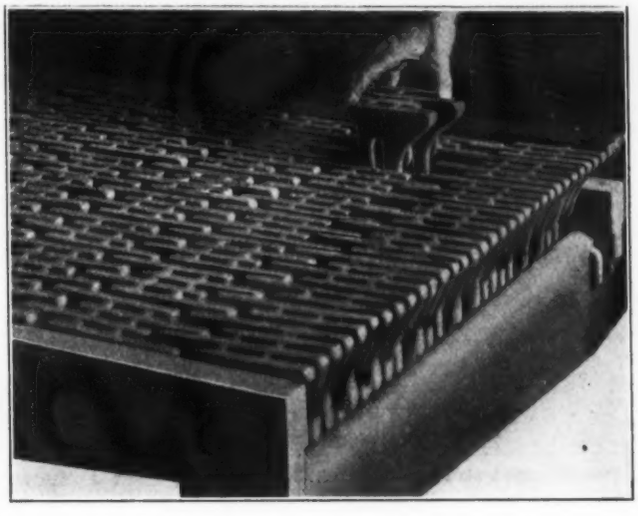
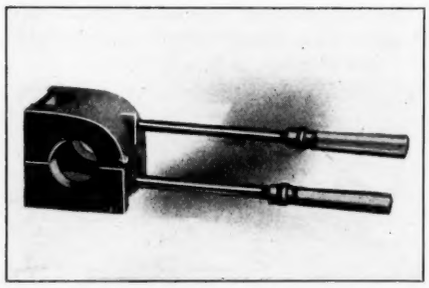


FIG. 2. SECTION OF COMPLETE GRATE

perfectly solid, and there is only a barely perceptible upset at the joint which can be machined off if desired.

The "Thermit" powder is put up in paper bags containing the quantity necessary to weld a given section, and is called a "welding portion." The size and price of the welding portion and equipment vary with the dimensions of the pipe to be welded. For instance, a welding portion costs from 35 cents to $2.75 for pipe welding; and from $1.35 to $6.10 for bars, according to whether round or



THE MOLD FOR THERMIT WELDING

square. Crucibles cost from $1.75 to $7; tongs, from $2 to $4.50; clamps, from $9 to $20. The cost of relining crucible shells ranges from 75 cents to $4. The clamps and tongs are permanent; the molds should last for sixty or more welds and the crucibles will withstand twelve to fifteen reactions. One-half teaspoonful of ignition powder is required to start a reaction, and one pound is sufficient for one hundred welds. The process is not applicable for pipes more than 5 inches in diameter, the prices quoted applying to pipe not exceeding 4 inches in diameter.

The "Thermit" process is controlled by the Goldschmidt Thermit Company, 90 West street, New York.

Substitute for the Brick Front Arch in Horizontal Boilers

This invention is designed to do away with the brick arch commonly used in boilers for closing off the passage between the fire-box and the front end of the boiler, and which deteriorates, on account of being subjected to intense heat, so that it usually has to be rebuilt several times during the life of the boiler. Therefore, instead of supporting the front end of the boiler by a heavy brick arch, the inventor proposes to substitute a coil of pipe *A*, extending back and forth across the end of the boiler and lying in a diagonal plane, the ends *B C*, Fig. 1, of this pipe entering the boiler adjacent to the lower side. This pipe may, if desired, extend to the outside and be connected to the feed-pipe *D*, thus acting as a feed-water heater. This coil is covered by a large sheet of asbestos *E*, Fig. 2, extending from the end of the boiler to the casing, and between the boiler and the side walls to a point midway of the hight of the boiler, so the hot gases from the fire cannot pass directly to the chimney, but must flow over the bridge to the opposite end and return through the tubes. The sheet of asbestos may be of any desired thickness, and arranged above the coil at a greater or less distance, or it may rest directly on the coil.

Additional coils *F* may be placed along the sides of the boiler, as shown in Fig. 2, supported preferably on water lugs *G* formed integral with the return bends of the coils, the opposite ends of the coils being connected to the boiler. Any suitable number of side coils may be provided.

As will be noticed, none of the coils lies directly between the fire and the boiler, so the heat they may absorb will not detract from that otherwise supplied to the boiler. The coils being in open communication with the boiler the water in the coils becomes heated to a very high temperature, and as this heated water is delivered to the boiler it is replaced by the boiler water of lower temperature. Furthermore, the coils cannot be burned out, as water circulates in them continuously. Jeremiah C. Parker, of Red Bank, N. J., is the inventor.

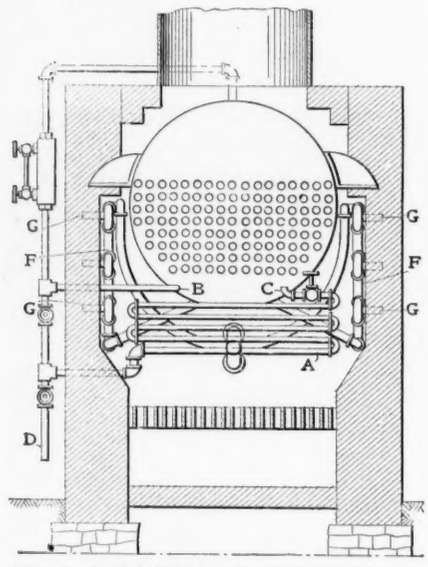


FIG. 1. CROSS-SECTION OF PARKER BOILER

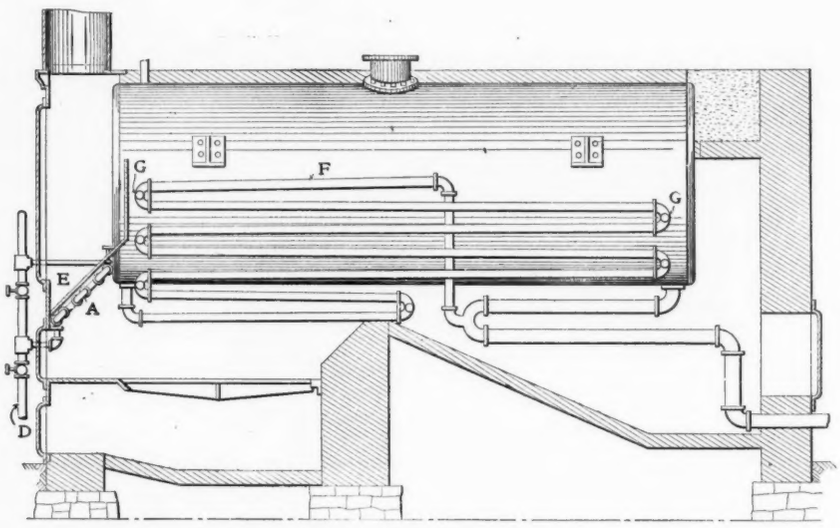
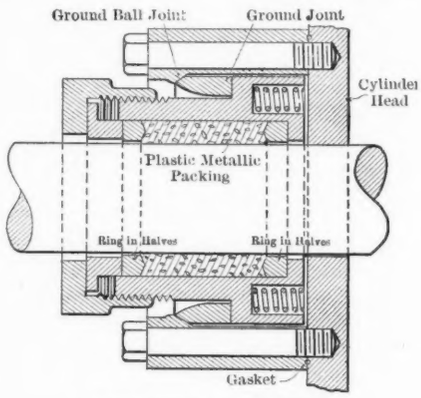


FIG. 2. SHOWING LOCATION OF COILS AND ASBESTOS SHEETING IN PARKER BOILER

A Vibrating Stuffing-box

We illustrate herewith a "vibrating" stuffing-box which has just been placed on the market. The feature of this device lies in its being designed to adjust itself automatically to any out-of-line movement of the piston-rod or valve stem. As will be noted, it is arranged within a casing and is held against a ground ball-joint ring by means of springs which, assisted by the steam pressure, keep the joints tight. Between the interior of the casing and the stuffing-box, and between it and the cylinder head, are clearances to permit the stuffing-box to move laterally so as to follow the out-of-line movement of the rod or stem, and by rocking on the curved face, or ball joint, to allow for angular movement. The spacing rings at each end of the packing



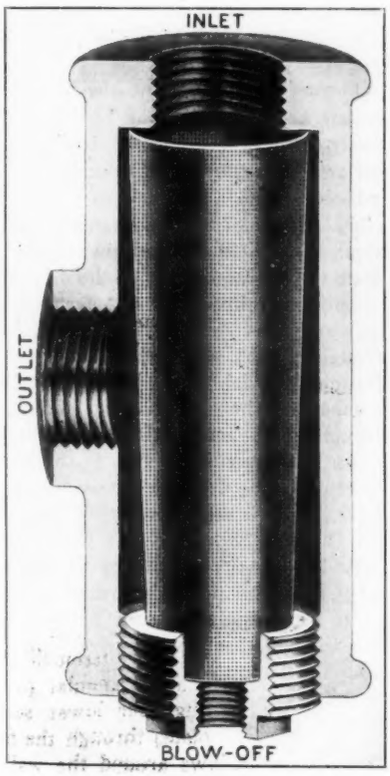
A "VIBRATING" STUFFING-BOX

act in conjunction with the latter to hold the stuffing-box out of contact with the moving parts.

It is stated that one of these stuffing-boxes was placed on the main piston-rod of a 100-horse-power engine, the rod being put out of line so as to provide an angular as well as a lateral movement of more than 1/16 inch. It is reported that the stuffing-box "floated" with the rod without resistance and did its work perfectly. In this instance the packing used was the "Safety" metallic packing made by the Steel Mill Packing Company, of Detroit, Mich., which also makes the "vibrating" stuffing-box. Any suitable packing can be used, however.

The Wright Filter

The Wright filter, which is shown in section herewith, was placed on the market only recently, although the makers have been experimenting with and applying it under practical observation for several years. It is so simple in construction as to need slight description, and it is claimed to be as practical as it is simple for eliminating foreign matter from water-pipe, boiler-feed and meter systems.



THE WRIGHT FILTER

The filtering tube is of brass with an area several times larger than the inlet. An ordinary blow-off valve screwed into the plug at the bottom, and to be opened occasionally, is provided to keep the device free from sediment. The filter tube can be easily removed by unscrewing the plug to which the tube is attached. This filter comes with inlet connections ranging from 1 to 3 inches, inclusively. It is made by the Wright Manufacturing Company, 55 to 59 Woodbridge street, Detroit, Mich.

The "Demon" Boiler-tube Cleaner

This device belongs to the rotary class and the principle embodied involves the operation, by direct water pressure, of three so-called folding pistons, which actuate the cleaner head very rapidly and with sufficient force to cut the scale on the inside of the boiler tube. There are two types of this cleaner, one for straight tubes and the other for curved. The essential features of the motor part, which are the same in both forms, may be studied in the sectional view Fig. 1.

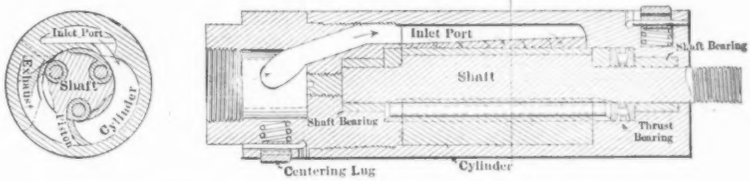


FIG. 1. SECTIONAL DIAGRAM OF "DEMON" BOILER-TUBE CLEANER

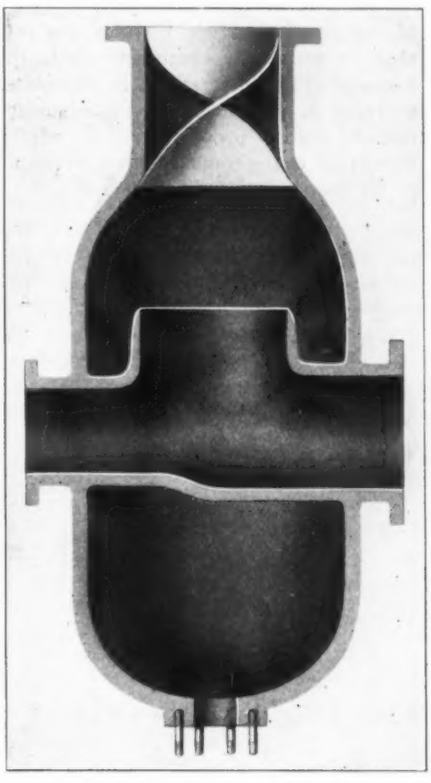
The barrel, or motor body, is provided with a longitudinal inlet port, designed so that the flow of the entering water is directed against the free end of each passing piston in such manner that the blades of the piston will be fully opened by the impact and, extending across the entire cylinder space, expose the maximum area to be acted upon by the water. The pistons are so located upon the shaft as to permit this action to take place as to each piston before the preceding piston reaches the exhaust, so that one piston is always in working position. Of course the motor has no dead center. The pistons being arranged lengthwise and parallel with the tube, the length of the tube can be taken advantage of to provide a piston having a very large working area. The cutting action of the cleaner head resembles that of a lathe tool. Front and rear centering lugs allow the cleaner to center itself, i.e., the cutter head is always in alinement with the tube, and adjustable cutter stops are employed to prevent damage to the tube itself. The parts work on roller bearings.

FIG. 2. "DEMON" CLEANER FOR CURVED TUBES

Fig. 2 shows the modified form of "Demon" cleaner as used in curved tubes. The flexible connection between the cutter head and motor, together with the short body, tapering slightly at each end, permits this form of cleaner to follow the sharpest bends ordinarily found in curved boiler tubes. These machines are manufactured by the General Specialty Company, 887 Niagara street, Buffalo, N. Y.

Swartwout "Vertical Angle" Separator

The accompanying view of the Swartwout "vertical angle" separator presents the newest form of separator which has been marketed by the Ohio Blower Company, of Cleveland, Ohio. It is designed for use in cases where steam is taken from the main line at an angle. It embodies the features of the Swartwout "vertical" and "horizontal" separators which were described in POWER in May and November, respectively, of last year.



THE SWARTWOUT ANGLE SEPARATOR

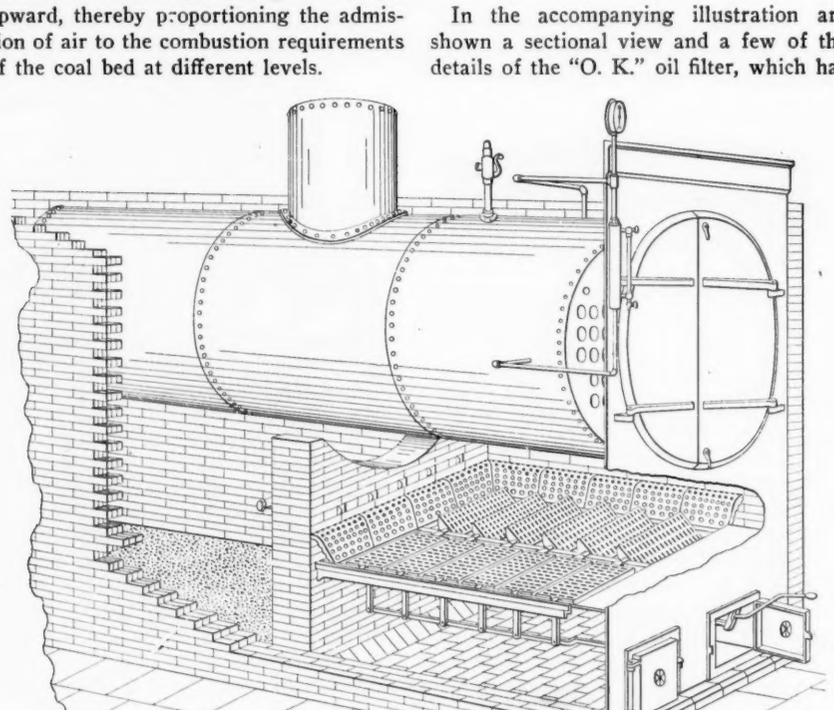
The principle involved in the Swartwout separators is the employment of a helical worm, cast integral with the throat of the device, which causes the passing

current of steam to rotate, throwing out the entrained water by centrifugal force. The water is swept down the internal walls of the separator to the bottom, and is trapped off. The dry center of the column passes straight through the vertical riser shown, and thence to the engine. There being no reversal nor sudden change of direction of flow, and the passages being ample as compared with the cross-section of the main, there is no appreciable loss of pressure in passing through the separator.

As the separator consists of a single casting, there are no parts to work loose and cause annoyance. It is built in sizes adapted to pipe from 1 to 12 inches in diameter.

The "Pittsburgh" Fuel-saving Furnace

In this boiler furnace the essential feature is the provision of means for facilitating the commingling of coal gas and air and for retarding the gases so as to enhance combustion. The means referred to comprise perforated cast-iron plates in the sides, and front of the bridge wall, and steam jets introduced through the bridge wall and projected over the body of the fire to produce a water gas that shall be practically consumed within the furnace. The perforations in the plates decrease in size from the grate surface upward, thereby proportioning the admission of air to the combustion requirements of the coal bed at different levels.



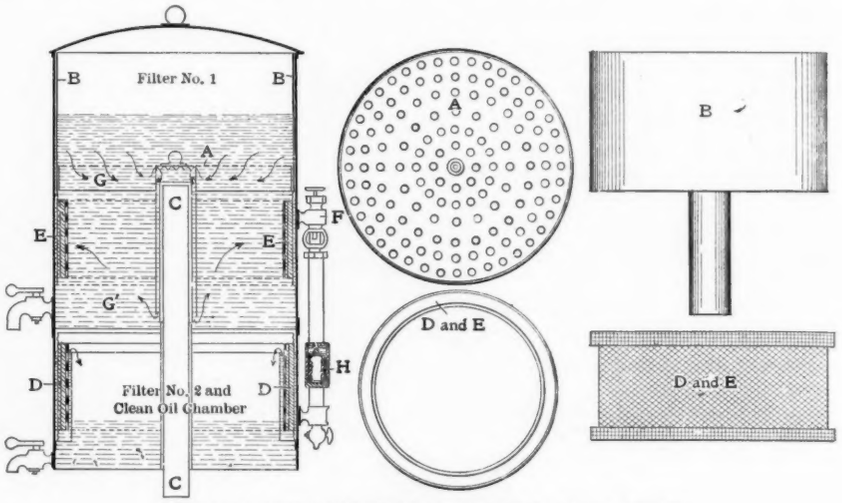
"PITTSBURGH" FUEL-SAVING FURNACE

With this grate slack can be used economically; the conditions requisite for complete combustion are produced; it protects the side sheets of a water-leg fire-box and the side walls of a brick-set boiler; facilitates the cleaning of fires, as every bar can be shaken and there are no "dead" bars to gather ashes and clinkers; and with ordinary care in firing it reduces smoke to a minimum. All the parts are easily and cheaply renewed.

This furnace is made by the Standard Car and Foundry Company, of Pittsburg, Pennsylvania.

The "O. K." Oil Filter

In the accompanying illustration are shown a sectional view and a few of the details of the "O. K." oil filter, which has been designed by Peter M. Knopp, a practical engineer, of Kansas City. This apparatus embodies two filter chambers, one above the other, so arranged that the oil passes through a series of wool-felt filters one in the form of a disk and the others



SECTIONAL VIEW AND DETAILS OF THE O. K. OIL FILTER

rings, filtering slowly enough to allow ample time for sediment to precipitate in the settling chambers *G G'*, special means being provided to heat the oil in passing.

The rate of filtering is under such complete control, Mr. Knopp states, that he has adjusted the device so that it filtered less than one quart in 24 hours without choking up, and in less than one second it was reset to work at the rate of 12 gallons in 24 hours.

Oil may be emptied into the filter from buckets or cans, or it may be fed continuously from an overhead separating tank, and two or more filters may be used in multiple.

The casing of the apparatus is cylindrical. In the illustration the arrows indicate the course of flow of the oil, which is continuous throughout. After it enters the top filter chamber it percolates through the disk *A*, which is covered with filtering material, into the upper settling chamber *G*, the outlet from which is through the corrugated perforations, shown in the sectional view, in the top of the central tube depending from the cylinder *B*, which is set in the upper filter section (marked "filter No. 1") in the manner indicated. From the top of the central tube the oil flows downward through a 1/16-inch annular passage between this tube and another tube surrounding it. From the bottom of the annular passage the oil rises through the lower settling chamber *G'*, percolates through the filtering ring *E*, follows around the walls of the filter to and through the angle sight-feed *F*, whence it flows with steady feed to the lower filter section marked "No. 2."

The arrangement of filter section No. 2 and the clean-oil chamber is such that every square inch of the filtering ring *D* is brought into active service; and in both filter sections the rings *D* and *E* are located around the outside of the filter chambers, so as to provide the maximum

surface for filtration. There is a float and needle-valve arrangement *H* in the connection between the filter sections to prevent automatically the overflow of filter section No. 2.

In the sectional assembly drawing *C* is a steam pipe, closed at the top, extending upward into the center of the apparatus, with an appreciable clearance between it and the wall of the tube surrounding it. Either high- or low-pressure steam, or hot water, may be used in the pipe for heating the oil as it passes downward through the annular clearance referred to. As the oil nearest the steam pipe flows in a thin sheet or film, very little heat is required to raise it to the desired temperature, say 140 or 150 degrees. Furthermore, since it does not come into direct contact with the steam pipe, there is no danger of the oil becoming too hot.

The filter is made by the O. K. Filter Company, of which Mr. Knopp is general manager, 2917 Woodland avenue, Kansas City, Mo.

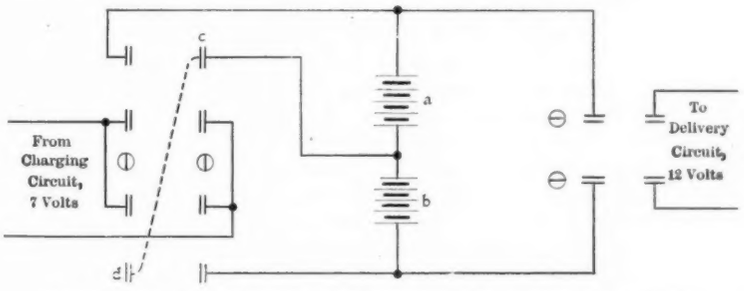
Inquiries

Questions are not answered unless they are of general interest and are accompanied by the name and address of the inquirer.

Charging and Discharging Storage Batteries at Different Voltages

Can I charge six cells of storage battery from a circuit giving 7 or 8 volts and at the same time take current from the battery at 12 volts? T. J. M.

Not exactly as you state it, but you can keep the battery fairly well charged from a 7- or 8-volt circuit while it is supplying current at 12 volts by charging one-half of the battery for a certain period of time and then charging the other half the same length of time. The time period should be relatively short in order to avoid unequal treatment of the two halves of the battery by reason of variation in its discharge rate. The accompanying diagram shows the necessary connections. A double-pole double-throw switch is connected between the battery and the charging circuit so that throwing the handle one way will connect one-half of the battery to the circuit and the reverse position will connect the other half. The battery cells are connected permanently in series; the switch jaws *c* and *d* are connected under the base, as indicated by the dotted line. Throwing the handle up puts the three cells *a* on charge; throwing it down puts the cells *b* on charge. The switch should be changed over at least every hour, punctually.

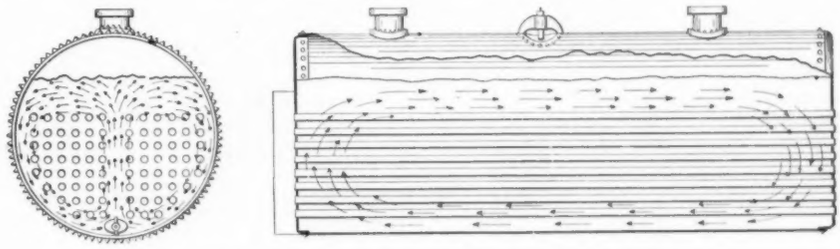


CONNECTIONS FOR CHARGING AND DISCHARGING A STORAGE BATTERY AT DIFFERING VOLTAGES

Circulation in a Horizontal Tubular Boiler

Will you please describe the circulation in a horizontal tubular boiler? A friend insists that it is up the sides and down the center. I claim that it is the reverse.

J. F. G.



CIRCULATION IN A HORIZONTAL TUBULAR BOILER

As shown in the accompanying sketch, the circulation in a horizontal tubular boiler is up the center and down the sides; lengthwise, it is from front to rear along the top, returning along the bottom from back to front.

Pumping-engine Performance in Different Units

A pumping engine performs 132,000,000 foot-pounds of work for every 100 pounds of coal furnished to it. (a) If the coal has a calorific value of 14,000 B.t.u., what percentage of the total energy of the coal is usefully employed? (b) If the coal consumption is 1300 pounds per hour, what is the horse-power developed by the pumping engine? (c) Assuming a gallon of water to weigh 8.3 pounds, how many gallons per hour can the engine raise through a hight of 113 feet? R. W. C.

The work per pound of coal is 1,320,000 foot-pounds and this is equivalent to

$$1{,}320{,}000 \div 778 = 1696.65$$

B.t.u. The proportion of heat energy usefully employed, therefore, is

$$\frac{1696.65}{14{,}000} = 0.12119,$$

or 12.119 per cent. The horse-power developed is equivalent to foot-pounds per minute divided by 33,000. At the rate of 1,320,000 foot-pounds per pound of coal and 1300 pounds of coal per hour, the foot-pounds per minute are equal to

$$\frac{1{,}320{,}000 \times 1300}{60} = 28{,}600{,}000,$$

and the horse-power developed will be

$$\frac{28{,}600{,}000}{33{,}000} = 866.67.$$

The foot-pounds per hour are 1,320,000 × 1300 = 1,716,000,000, and this is the equivalent of raising 15,185,840 pounds 113 feet high in one hour; consequently, that rate of working will raise 15,185,840 ÷ 8.3 = 1,829,619.28 gallons of water per hour 113 feet high.

Kilovolt-amperes and Kilowatts

What is meant by kilovolt-amperes and what is the relation between them and kilowatts?

G .W. K.

Kilo means one thousand. Volt-amperes are volts × amperes; hence one kilovolt-ampere is equal to one thousand volt-amperes. In a direct-current circuit volt-amperes are identical with watts, and kilovolt-amperes are therefore the same as kilowatts. In alternating-current work, however, the electromotive force and current do not always rise and fall in unison and when this is the case, volt-amperes are not identical with watts; watts are the product of volts × amperes × power factor, while volt-amperes are the product of volts × amperes, as before. In short, kilowatts = kilovolt-amperes × power factor, and the power factor of a direct-current circuit is always 1, while that of an alternating-current circuit may be 1 but is usually less.

Energy, Work and Power

Please explain just how energy, work and power differ from each other.

D. X. S.

Energy is the ability to do work; power is the rate of doing work; work is the expenditure of energy. The steam in a boiler contains a certain amount of energy or ability to do work; when it is admitted to an engine cylinder it does work in forcing the piston back and forth, and the power developed by the engine is the rate at which the work is done.

Book Reviews

COMBUSTION AND SMOKELESS FURNACES. By Joseph W. Hays. The Hill Publishing Company, New York. Cloth, 104 pages, 6x9; illustrated. Price, $1.50.

The author being a resident of Chicago has lived with the smoke problem, and for a number of years has made it a special study. His book is designed to meet the needs of those most interested in the suppression of the smoke nuisance—owners and engineers of power plants. To this end the reader is told, in a way that will be readily intelligible to the operating engineer and the layman, enough of the principles of combustion to illuminate the problem and to show what is possible of attainment with commercially practicable amounts of effort and expense. The book is entirely free from bias in favor of any particular system or apparatus; the desirable features and the faults of various types of smokeless furnaces being pointed out, the essentials for success enumerated, possibilities of damage to boiler and setting and impairment of efficiency indicated. The author arraigns the evaporative test as an evidence of furnace efficiency. Such tests, he says, "may be made to prove anything desired; place no confidence in them. The above stricture is not intended to apply to the evaporation test as an evidence of boiler efficiency, properly and honestly made; the evaporation test is indicative of something as to furnace efficiency, but the best of such tests is not conclusive. The proper test to apply to any device claiming to improve combustion is a flue-gas analysis. The evaporation test is necessary to determine boiler efficiency, but should not be employed to determine anything related to combustion."

POWER TRANSMITTING MACHINERY. George V. Cresson Company, Philadelphia, Penn. 350 pages; 6x9 inches; cloth; illustrated.

This is a satisfactory example of what a trade catalog ought to be. The ordinary builder of a catalog for this firm would have set forth in laboriously evolved paragraphs that their shafting was round, straight and well polished, that their pulleys were well balanced and made of iron or steel when not otherwise directed, etc.; together with yards of description of plant and facilities. This catalog tells all that a prospective customer wants to or needs to know of the size, history and reputation of the works in one bird's-eye view of the works as a frontispiece, and the simple inscription: Established 1859. Incorporated 1892.

Pages 4 to 17 are devoted to information on shafting, not Cresson's shafting, but Cresson's information, engineering formulas and notes derived from or proven out by the experience of the engineers of the company, aiding the designer to determine the best size of shafting to order for a given service and for given conditions, with tables of yellow pine stringers, footing pieces and bolts for erecting the same. Belting, and the safe speed of cast-iron wheels are treated, and tables of areas and circumferences of circles, polygonal angles, square roots of ratios, decimal equivalents of parts of an inch and of a foot and of dimensions of standard, special and reducing keys are given.

Coming to the treatment of their product, the first thing that strikes one's eye on opening to a page of the catalog proper is the legend: List No. 000, in black-face type nearly half an inch in hight. No word is said about it, but the presumption is that the list number will be included in making up the order. A code is furnished in the latter pages corresponding to these list numbers for use in telegraphic orders.

The author of the catalog has evidently worked over a drafting board, for in addition to the half-tone engraving, printed in two colors, of each subject treated, an outline sketch is furnished showing it as it would be represented in a drawing with a table of dimensions for the various sizes corresponding to lettered dimension lines upon the drawings. The book will be a veritable prize to a designer and, as we started in by saying, is an excellent example of the ideal way to put one's product before the people who are to use it.

ENGINEERING IN THE UNITED STATES. By Frank Foster. Published by the University Press, Manchester, England. Cloth; 106 pages exclusive of index, 5¾x9⅛ inches. Price, 75 cents.

This is a report on the results of a twelve months' tour in America by the author, who held a Gartside scholarship at the University of Manchester. On the whole it is a much fairer summing up or comparison of industrial conditions in Great Britain and America than one is accustomed to look for from a British viewpoint. It is apparently without bias, and the impression the reader gets of the author is very pleasing. Mechanics, especially, can find a great deal of instructive material in this book. The comments on high wages, high costs of living and high tariff are interesting, particularly the latter, although no plan is presented for "tariff reform." Protection and unemployment, protection and wages, trade unions and apprentices, trade unions and unemployment, are handled with discernment and some good suggestions offered for the equalization of differences existing because of an apparent misunderstanding of the true co-relation of these subjects. Educational institutions, street railways, specialization, organization, as well as engineering in several of its practical branches are among the things treated of, the whole being "dished up" entertainingly. Mr. Foster visited a number of large electrical and mechanical industries here, and spent some time as an employee in manufacturing works and in a power station.

PEABODY ATLAS—SHIPPING MINES AND COAL RAILROADS OF THE CENTRAL COMMERCIAL DISTRICT OF THE UNITED STATES, ACCOMPANIED BY CHEMICAL, GEOLOGICAL AND ENGINEERING DATA. By A. Bement. Published by the Peabody Coal Company, Chicago, Ill., 1906. Cloth, 149 pages, 16¾x18 inches; illustrated. Price, $5.

This work is one of undeniable excellence, and it is easy to see that it cost a great deal more to publish than the price charged for it. It covers the coalfields of Illinois, Indiana, Ohio, Michigan, Iowa, Missouri and western Kentucky. Besides an adequate number of colored maps, which clearly designate the localities, and the chief features of each, the work contains a large amount of technical information concerning coal, including a brief description of the geology of each coal field, detailed treatment of the various coal seams, and their chemical analyses. Attention is given to the management of fires and the determination of conditions of combustion by chemical analysis of chimney gases is treated of; also the essential instruments for analytical purposes are illustrated and described. An interesting chapter is devoted to several types of smokeless furnaces, which are also illustrated.

THE ENGINEERING INDEX, Vol. IV, 1901-1905. *The Engineering Magazine*, New York. Cloth, 1, 2, 3, 4 pages, 6¼x9 inches. Price, $7.50.

This fourth volume of "The Engineering Index" represents the continuation of the work originally started by the late Prof. J. B. Johnson in the *Journal* of the Association of Engineering Societies in 1884, and turned over by that association to *The Engineering Magazine* at the close of 1895. In it the articles appearing in 250 technical and engineering journals in six different languages are indexed with brief abstracts indicating the length and character of the articles. Only articles of permanent value are included and the volume serves as a guide to the existing data and information upon any particular branch of the field covered. The present volume contains upward of 50,000 entries, as against some 40,000 in the preceding volume. The book constitutes a master key to the mass of engineering information locked up in engineering libraries and files of technical publications and will be of immense service to searchers for particular information in the field which it covers.

The Kern River Power Plants

On page 167 of the March number was published an item, relating to the 60,000-volt Kern river power plant, in which it was stated that the equipment of the Los Angeles Edison Company's Kern river plant was built by the Allis-Chalmers Company. This was an error in the Allis-Chalmers Company's news bulletin from which we derived our information. The Allis-Chalmers generators referred to in the article in March were installed in the Kern river plant of the Pacific Light and Power Company. The equipment in the Edison Company's high-potential plant on the Kern river was built by the General Electric Company and comprises four three-phase 5000-kilowatt alternators which will deliver current at 50 cycles and 2300 volts to the step-up transformers, which are provided with taps giving several line potentials ranging from 33,000 to 75,000 volts. The primary windings of the step-up transformers are delta-connected, and the secondary or high-tension windings are Y-connected.

Summer School at the University of Wisconsin

The seventh annual session of the Summer School for Artisans, held under the direction of the College of Engineering of the University of Wisconsin, begins June 24, and continues for a period of six weeks.

Courses of study are offered in the following subjects:

Engines and Boilers

Lectures and laboratory courses covering the theory, construction, management and testing of steam engines, boilers, gas engines and gas producers, refrigerating machines, etc.

Applied Electricity

Lectures and laboratory courses covering the theory of direct- and alternating-current dynamos and motors, the operation and method of testing electrical machinery, batteries, transformers and other apparatus, photometry, and calibration of instruments.

Mechanical Drawing and Machine Design

Elements of applied mathematics, courses in mechanical drawing and machine design adapted to the preparation of the students.

Materials of Construction, Fuels and Lubricants

Lectures on the properties of materials accompanied by laboratory tests; lectures on fuels and lubricants with laboratory tests on the heating value of coals and efficiency of lubricants.

Shop Work

Practice with hand tools, wood and metal-working machinery, and in blacksmithing and pattern making.

Manual Training

Lectures and laboratory courses adapted to the requirements of manual training teachers.

The entire laboratory and shop equipment belonging to the College of Engineering is used by the students in the summer school.

The requirements for admission do not extend beyond a working knowledge of English and arithmetic, but the policy is to allow a large amount of individual work so that the student may take advantage of all the preparation he has obtained. A bulletin describing the work of the School for Artisans in detail will be sent on application to Frederick E. Turneaure, Dean, College of Engineering, Madison, Wis.

New Power House Wrecked by Heavy Storm

During the afternoon of April 7 a heavy wind and rain storm practically wrecked the new power house under construction for the Cincinnati Northern Traction Company, at Hamilton, O. The roof and side walls were demolished, and more or less damage resulted to the turbines and other machinery that were being installed. The total loss, estimated at $100,000, falls upon the contractor.

Electricity at Jamestown

Electricity for power and illumination at the Jamestown Exposition is generated by Curtis steam turbines in the power house of the Norfolk Railway and Light Company, seven miles away, and transmitted to a model sub-station in Machinery Hall. Thus this is the first Exposition at which electric power will be generated by steam turbines. The turbines and the entire electrical equipment is furnished by the General Electric Company. All the electrical machinery follows standard lines similar to that installed at the St. Louis, Pan-American and other Expositions. It is expected that the illuminating features will excel any previous display.

On April 2 the large fly-wheel in the main building of the Medway (Mass.) Box Company's plant "burst" and heavily damaged the plant. The primary cause is unknown. Nobody was injured.

Personal

James O. Westberg has been appointed manager of the stationary department of the Bird-Archer Company, with headquarters at the main offices of the company in New York.

W. H. Booth, of England, whose contributions are familiar to Power readers, is in the United States for a brief visit, and while in New York may be reached by addressing him care of Power.

A. B. Crozier recently severed his connection with the Cudahy Packing Company's mechanical department, and opened an office as consulting mechanical engineer at 510 Kansas City Life Building, Kansas City, Mo., where he is associated with Robert C. Barnett, civil engineer.

W. R. Jennison, Prudential Building, Atlanta, Ga., southern representative for many years of the Lane & Bodley Co., Cincinnati, has severed his connection with that company and will hereafter devote his attention exclusively to the sale of Wheeler condensers, Weber concrete steel stacks and Cochrane feed-water heaters and separators.

Professors William E. Geyer and John Burkitt Webb will retire from the faculty of the Stevens Institute of Technology, Hoboken, N. J., at the end of the school year in June. Professor Geyer, who joined the faculty in 1870, is head of the physics course. Professor Webb occupies the chair of mathematics and mechanics. He has been at Stevens twenty-one years; he has invented many devices. Both will be retired at their own request.

Obituary

Col. Simeon J. Fox, president of the National Pipe Bending Company, died suddenly of heart disease, on March 25, at his home in New Haven, Conn. He was 65 years old, and had lived in New Haven more than sixty years, becoming prominently identified with State and city government, was a member of the Union League Club and of Admiral Foote Post, G. A. R. His connection with the National Pipe Bending Company began in 1882. One of deceased's three sons, Edward L. Fox, is secretary and treasurer of the company.

Business Items

The New York branch of the Green Fuel Economizer Company, has removed to the new West Street Building, 90 West street. The new offices will be considerably larger than the old, in order to take care of the increasing business in economizers, fans, blowers and exhausters. Since entering the fan business,

this company has built many large fans for mechanical draft, including several with overhung wheels, larger than any of this type ever built before. The Green Fuel Economizer Company's New York representative is William Downs, who has occupied the old offices at No. 74 Cortlandt street since 1889.

The Bird-Archer Company, on May 1, will remove to its new offices on the twelfth floor of the West Street Building, 90 West street, New York City.

The Sims Company, Erie, Penn., has received an order from the Pittsburg & Lake Erie Railroad Company for a 3500-horse-power feed-water heater.

The Heine Safety Boiler Company has appointed N. H. Emmons manager of its Boston office, in place of W. E. Muse, who has resigned on account of ill health.

Hunt, Mirk & Co., San Francisco, Cal., have been appointed Pacific coast agents of the American Engine Company for the sale of American-Ball engines and electrical machinery.

By a decision of the court the suit brought by the Consolidated Engine Stop Company against the Locke Regulator Company, for infringement of patent on engine stop, has been dismissed.

The Ball Engine Company, Erie, Penn., builder of automatic and Corliss engines, has opened a branch office at 1213 Chemical Building, St. Louis, under the management of O. L. Collins.

The Allis-Chalmers Company has received orders for six large electric generators for various services, two are for the new steel plant of Jones & Laughlin Steel Company, at Aliquippa, Penn.

Gardner Cornett, of Providence, R. I., has been recently elected vice-president of the American Steam Gauge and Valve Manufacturing Company, his office being located at the headquarters of the company, 220 Camden street, Boston.

The Goldschmidt Thermit Company has removed its main offices to the fourteenth floor of the West Street Building, 90 West street, New York. This company has opened a Pacific coast branch at 432 Folsom street, San Francisco, Cal., under the management of L Heynemann.

The Ball Engine Company, Erie, Penn., announces that it has opened a branch office at 39 Cortlandt street, New York, for the sale of its automatic and Corliss engines. This office will be under the management of Lancelot Copleston, who is well known to the New York trade.

The new electrical engineering building of Worcester Polytechnic Institute, which is to be completed this fall, and which will be the largest one of its kind in the country, costing over $125,000, is to be equipped with a complete heating and ventilating outfit supplied by the Green Fuel Economizer Company, of Matteawan, N. Y.

The Smith Gas Power Company, Lexington, Ohio, reports recent sales of 1980 horse-power of Smith automatic suction gas producers. Two of the orders included in this total are especially interesting in view of the fact that they are second orders from the Franklin Steel Works, Cambridge, Mass., and the Harvey Spring Company, Racine, Wis., which have been using these producers for some time.

The Alberger Pump Company, with offices at 95 Liberty street, New York, has been organized to manufacture and sell centrifugal and turbine pumping machinery embodying novel and important improvements. The officers of the new company will be: President, Louis R. Alberger; vice-president, George Q. Palmer; secretary and treasurer, B. W. Pierson; chief engineer, Frederick Ray; first assistant, William S. Doran.

The Joseph Dixon Crucible Company, of Jersey City, N. J., has issued an interesting pamphlet of twenty pages on the subject of "Air Compressor Lubrication," copies of which will be sent free on request. The pamphlet deals with the difficulties and dangers of oil lubrication in this class of work, gives instances of destructive explosion of oil in the hot compressed air, and shows how graphite may be used either dry or with oil or soapsuds for the purpose of lubrication.

Adam Cook's Sons, 313 West street, New York City, the makers of Albany grease, have received a letter from C. W. Sexton, superintendent of the Waycross Electric Light and Power Company, Waycross, Ga., in which he says: "We have given your Albany grease a good fair trial and have found it the best grease we ever used. One No. 2 compression cup full of Albany grease lasted for three weeks continuous running on the crank-pin of our 18x42 Corliss ice machine engine, running 65 revolutions per minute."

The central heating and power plant to be completed during the coming summer for the Capitol buildings at Washington will contain 9000 horse-power of water-tube boilers. These boilers will be equipped with Foster superheaters, manufactured by the Power Specialty Company, 111 Broadway, New York City. At the recent opening of bids on boiler, stoker and superheater equipment for the above plant twenty-two of the tenders, including that of the successful bidder, offered the above superheater; the two remaining boiler companies bidding offered a bare-tube type of superheater.

The Casualty Company of America has recently added to its other lines of insurance that of engine and fly-wheel inspection and insurance. The remarkable success of this company in steam-boiler insurance, having increased the premium writings on its books from $48,000 to $329,000 in less than three years, would indicate that the new venture will be most successful. The company is issuing a new contract on different lines from any heretofore attempted, covering not only for damage to property and loss of life, but also for daily loss of profits and income, as the result of the shutting down of machinery consequent upon an explosion.

The B. F. Sturtevant Company, of Boston, Mass., has just received from the Interborough Rapid Transit Company, of New York, a repeat order for standard economizers. The previous order was for the equipment of 26,000 boiler horse-power. The total number of tubes to complete both orders will be 11,200, making approximately 100 carloads. Other orders for economizers were recently received from the Brooklyn Rapid Transit Company, New York Edison Company, Uncle Sam Portland Cement Company, Greenwood Bros. & Co., and Cochrane Manufacturing Company. The Sturtevant company has also secured recent orders for seven generating sets and three mechanical-draft equipments.

The increasing trade of the mechanical department of the Minneapolis Steel and Machinery Company has made it imperative for them to add another large building of steel and brick construction to their already large plant. It was only a few months ago that the company had to double the capacity of the foundry. When the new building is completed, it will have a total of over 12 acres of floor space within a yard area of over 20 acres. A company of Minneapolis men has given to that city an enterprise which is adding a new chapter to Minneapolis fame in all parts of the country. In the short period of a year, since they have secured the right from Germany to build the Muenzel gas engine and suction gas producer, they have installed over thirty plants and have orders in the shops for a great many more.

On May 1 the Chicago branch of Niles-Bement-Pond Company, the largest manufacturer of machine tools in the world, will occupy its new offices on the sixth floor of the new Commercial National Bank Building, Clark and Adams streets, Chicago. In this building will be located also many of the large engineering and steel companies. The Pratt & Whitney Company will abandon its showroom and offices at 46-48 South Canal street, and will combine its machinery sales department with that of the Niles-Bement-Pond Company The showroom and stock of Pratt & Whitney small tools, and the small tools sales department, will be located on the ground floor of the new Plamondon building, Clinton and Monroe streets, where a complete line of Pratt & Whitney small tools and gages will be carried in stock. Geo. M. Mills, who for several years has looked after the interests of these companies in the Chicago territory, will continue as manager of the Chicago offices.

What is said to be the largest order for water-tube boilers ever let by the United States Government was awarded during the closing hours of Congress to the Atlas Engine Works, of Indianapolis. It is for the boiler equipment of the new central power plant located in Garfield Park, Washington, which when completed is to furnish heat, light and power for the three buildings surrounding the Capitol—the Congressional Library, the new Senate and House office buildings and the Capitol itself. Sixteen high-pressure Atlas water-tube boilers of approximately 600 horse-power each were purchased. They will be erected in batteries of two and will be equipped with Roney stokers and Foster superheaters, necessary because of the exceptionally high-degree superheat called for in the turbines to be used. Deliveries will start on August 1, and it is anticipated that the plant will be ready for operation by the first of next year. J. G. White & Co. are the Government's consulting engineers on the work.

After many experiments the Keystone Lubricating Company, of Philadelphia, advises us that it has produced a lubricant that "stays put," the product being the Keystone grease. It is claimed that not only does this grease embody all the essential qualities of the highest form of lubricant, but it is absolutely pure. There is nothing in it but concentrated fine oil. It spreads and stays spread. It does not leak and is not "messy." There is no condition in which it becomes rancid, it is absolutely pure, containing no acids, grit, resinous oils, or any substance having a tendency to affect any kind of metal. Nor does it contain ingredients of a gummy nature. The Keystone Lubricating Company guarantees that one pound of its grease will outlast three to four pounds of any other grease, or four to six gallons of any lubricating oil, and to any engineer who will test Keystone grease the company will send one barrel and eight brass cups free on ninety days' trial, it being understood that after trial if the grease is not entirely satisfactory whatever quantity remains is to be returned with cups, the Keystone Lubricating Company paying shipments both ways.

New Catalogs

Alberger Condenser Company, Catalog No. 7. Cooling towers. Illustrated, 32 pages, 6x9 inches.

Pilley Packing and Flue Brush Manufacturing Company, 606-610 South Third street,

St. Louis, Mo. Catalog No. 9. Packings, flue brushes, flue scrapers, etc. Illustrated, 80 pages, 3½x6 inches.

Charles Engelhard, 41 Cortlandt street, New York. Catalog. Heraeus "Le Chatelier' pyrometer. Illustrated, 55 pages, 6x8 inches.

Robins Conveying Belt Company, 15 Park Row, New York. Bulletin No. 17. Conveying and elevating machinery in pulp and paper mills. Illustrated, 16 pages, 6x9 inches.

McNab & Harlin Manufacturing Company, 50-56 John street, New York. 1907 Catalog. Brass and iron goods for steam, water and gas, valves, cocks, fittings, etc. Illustrated, 285 pages, 5x7½ inches.

Garvin Machine Company, Spring and Varick streets, New York. Circular No. 53. Vertical spindle milling machines. Illustrated. Circular No. 54. Motor driven milling machines. Illustrated.

Morris Metallic Packing Company, Inc., Eleventh and Ridge avenue, Philadelphia, Penn. Booklet. Metallic packing for piston rods and rotary valve steam packing. Illustrated, 19 pages, 3x6 inches.

General Electric Company, Schenectady, N. Y. Bulletin No. 4485. Portable gasolene engine and generator outfits. Illustrated, 4 pages, 8x10½ inches. Bulletin No. 4487. Small plant direct current switchboard panels. Illustrated, 6 pages, 8x10½ inches.

Jeffrey Manufacturing Company, Columbus, Ohio. Bulletin B. Grab bucket system and coal and ash handling machinery for power houses. Illustrated, 12 pages, 6x9 inches. Bulletin C. Coal and ashes handling machinery for power plants. Illustrated, 16 pages, 6x9 inches.

Allis-Chalmers Company, Milwaukee, Wis. Bulletin No 1045. Rotary converters. Illustrated, 16 pages, 8x10½ inches. Bulletin No. 1510. Direct connected Reynolds Corliss engines. Illustrated, 12 pages, 8x10½ inches. Bulletin No. 1501. Belted Corliss engines. Illustrated, 20 pages, 8x10½ inches.

Ingersoll-Rand Company, 11 Broadway, New York. Catalog. No. 91. Davis Calyx diamondless core drill. Illustrated, 62 pages, 6x9 inches. Catalog No. 36. Ingersoll-Sergeant air and gas compressors. Illustrated, 183 pages, 6x9 inches. Form 45A. Rock drills. Illustrated, 3½x6 inches.

Allis-Chalmers Company, Milwaukee, Wis. Bulletin No. 1417. Smelting furnaces and accessory equipment. Illustrated, 47 pages, 8x10½ inches. Bulletin No. 1508. Christensen straight air-brake equipments. Illustrated. 12 pages, 8x10½ inches. Bulletin No. 1506. Electric hoists. Illustrated, 16 pages, 8x10½ inches.

Ingersoll-Rand Company, 11 Broadway, New York. Catalog H36. Class "H" air and gas compressors. Illustrated, 64 pages, 6x9 inches. Pamphlet, Form 410. Some economical applications of compressed air in cotton industries. Illustrated, 11 pages, 6x9 inches. Catalog No. 46. Rand rock drills. Illustrated, 84 pages, 6x9 inches.

Help Wanted

Advertisements under this heaa are inserted for 50 cents per line. About six words make a line.

WANTED—Experienced draftsman for stoker design. State experience and salary expected. Box 217, POWER.

WANTED—Man thoroughly understanding the manufacture of liquid air on a large scale; fine opportunity; also to purchase liquid air plant. Address Frank Shuman, Tacony, Philadelphia, Pa.

WANTED—Draftsman experienced on power installations; must be rapid and accurate; permanent position with excellent opportunity for right man; state references, age and salary required. Box 230, POWER.

WANTED in the cities of Pittsburg, Detroit, New Orleans, Birmingham, Ala., San Francisco and Cincinnati, jobbers or manufacturers' agents on commission to handle a full line of steam gages and steam specialties. "X. Y. Z.," Box 228, POWER.

WANTED—Steam and gas engine detailers, and experienced tracers for machine details; exceptional opportunity for brainy men; state age, experience and salary expected. Address Employment Manager, Minneapolis Steel and Machinery Co., Minneapolis, Minn.

WANTED—Steam engineer for railroad company; must have thorough experience in design and erection of modern power stations of large capacity; location, New York; salary, $3000. Address application, giving education, experience, etc., Box 231, POWER.

WANTED—An engineer to take charge of our steam plant May 1; engine is 12x22x30 tandem compound; man must thoroughly understand using indicator and have one; work is from noon until midnight; wages $2.75 per day with steady work. Oakes Milling Co., Oakes, N. Dak.

Situations Wanted

Advertisements under this head are inserted for 25 cents per line. About six words make a line.

MECHANICAL ENGINEER traveling in Europe this summer wants to correspond with parties desiring informations, reports, representation, etc., of technical nature; has successful experience in this work. Box 225, POWER.

POSITION WANTED—Thoroughly practical and technical engineer with long experience in operating and management of steam, water and electric power plants desires position to represent good manufacturing or contracting company. Box 226, POWER.

CHIEF ENGINEER wants position; experienced on portland cement, street railway and manufacturing plants of 5000 h.p.; up to date on heavy compound engines, compressors, steam turbines, water tube boilers, mechanical stokers, electric generators, motors, good machinist, makes all repairs, licensed, accustomed to overalls. Box 232, POWER.

Mechanical engineer with business, executive and organizing abilities at present in leading position with concern manufacturing successful gas producers and engines desires to change; can furnish own new designs in above as well as pumps, compressors, and power transmission machinery. Location and line immaterial. Box 224, POWER.

SITUATION WANTED by engineer; (35); abstemious; married; can make own repairs; never failed to effect saving in fuel where I have had charge; experienced with A.C. and D.C. generators, and all classes of steam engines; owners of plants who are not getting what they should out of fuel and supplies should address, Lock Box 336, Rogers, Ark.

WANTED—By a practical mechanic; position as chief engineer in a power plant; 20 years' practical experience with modern machinery, or a position as assistant master mechanic, or assistant superintendent, in engine-building shop; thoroughly practical and reliable and up to date; have large amount of tools and instruments. Address "B. E.," Box 223, POWER.

EXPORT MANAGER at present treasurer of large export company would consider change for personal reasons; thoroughly experienced in foreign credits and all matters relating to exporting; large acquaintance among foreign importers and qualified to make good; salary, $5000, or part salary and contingent interest. Address "C. W.," Box 227, POWER.

WANTED—POSITION—After May 1, I will be at liberty to accept position as manager, or chief engineer, where over twenty years of practical experience in handling all kinds of engines, A.C. and D.C. generators (also the public) counts; own my own test instruments; best of reference; am practical, also technical graduate of corresponding school in electrical engineering course. Address "Diesel," Box 229, POWER.

Miscellaneous

Advertisements under this head are inserted for 50 cents per line. About six words make a line.

ASBESTOS AND SHEET iron laggings for cylinders and steam chests; brass or nickel bands. John E. Widen, 87 So. Elliott Place, Brooklyn, New York.

IF YOU DESIRE to learn the latest improvement in steam boilers correspond with the Detroit Water Tube Boiler Co. See their advertisement on page 158.

ENGINES AND BOILERS, ⅛ to 2-h.p. engine castings in sets. Models and general machine work. Sipp Electric and Machine Co., Paterson, N. J. Catalog 4c.

WANTED—Back volumes of POWER, or a set from Vol. 1 to 1900 inclusive; offers of any single numbers or volumes acceptable. Address P. O. Box 1075, St. Louis, Mo.

WANTED—Agents to carry a staple article in steam fittings; now supplied by but one company in the U. S.; liberal discounts to salesmen and agents. Box 234, POWER.

PATENTS—H. W. T. Jenner, patent attorney and mechanical expert, 608 F St., Washington, D. C. I make free examination and report if patent can be had, and exact cost.

THE BIGGEST AMOUNT of Know-How for the money may be had by sending 8 cents for my catalog; full of practical information. P. M. Knopp, 2917 Woodland Ave., Kansas City, Mo.

C. L. PARKER, solicitor of patents, 604 F St., Washington, D. C. Patents secured promptly and with special regard to the legal protection of the invention. Handbook for inventors sent upon request.

ENGINEERS AND FIREMEN—Send 10 cents in stamps for a 40-page pamphlet containing a list of questions asked by an examining board of engineers. Stromberg Publishing Co., 2703 Cass avenue, St. Louis, Mo.

For Rent

LARGE FLOORS for rent, 60x210 ft. in thriving country town near Boston and Providence. Steam and electric power, light and heat. Railroad switch. Box 222, POWER.

For Sale

Advertisements under this head are inserted for 50 cents per line. About six words make a line.

HOPPES LIVE STEAM PURIFIER, 750 H.P.; good as new; immediate delivery; bargain. Box 233, POWER.

FOR SALE—One 25 h.p. Allfree steam engine, little used and perfect condition. Crist, Scott & Parshall, Cooperstown, N. Y.

FOR SALE—Cyclopedia of electricity; five volumes, three-quarters red morocco, leather binding; books are in fine condition; used but short time; cost $19; will sell for $9. Address "R. S. M.," POWER.

FOR SALE—A slightly used engineering library. Steam, electrical, mechanical. Twelve volumes bound in three quarters red morocco leather. Cost new, $60 Will sell for $18. Address "R. L. N., POWER.

FOR SALE—Cyclopedia of engineering; steam, electrical, mechanical; bound in three quarters morocco; four volumes; cost new, $19; are in fine condition; will sell for $9. Address "M. B. T.," POWER.

FOR SALE CHEAP—Latest type 75 H.P. Otto producer gas engine and equipment complete; used eighteen months; delivery sixty days; can be seen running at our plant. Rushmore Dynamo Works, Plainfield, N. J.

FOR SALE—Factory site of 57 acres in Wilmington, Del., within ten minutes walk from Pennsylvania depot, three railroad connections, 1500 feet of water front on navigable river; no city taxes for ten years. Address E. Krell, Narberth, Pa.

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How to Set a Riding Cut-off Valve

The Principal Points which Engineers Ought to Know About the Action of the Well-known Meyer Valve

BY HUBERT E. COLLINS

This article has to deal with what is known as the Meyer cut-off valve, of which there are in use a few different modifications of design, such as: (a) Single valve and cut-off set over the ports in a mid-position between the ends of the cylinder; (b) valves bridging ports at each end of the cylinder operating as one; (c) separate valves over each steam valve port, and separate valves over the steam and exhaust ports.

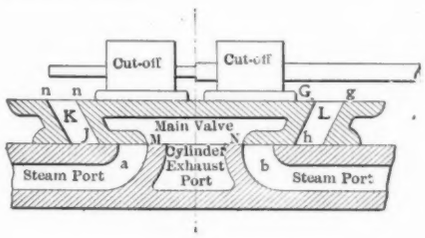


FIG. 1

The cut-off valve operating in conjunction with the main steam valve, or valves, may be riding over or inside cut-off, but the action must be the same in principle. The only difference is that with more valves used a shorter steam port and independent and closer adjustment of action is secured.

It is the purpose of this article to treat only of the simplest of Meyer cut-off valves, of which Fig. 1 is an illustration. This valve may be so constructed and connected that it may cut off the steam at a fixed and unvarying point, or by means of a hand wheel vary that point, to offset variations in boiler pressure, as in marine work; or where regularity of speed is essential, the cut-off valve may be operated from a shaft or other governor, the main valve being operated from a fixed eccentric.

In the case under discussion the main valve and the cut-off valve are each driven by a separate fixed eccentric. The resemblance of the Meyer cut-off valve, in this connection, to the D slide valve is shown by comparing Figs. 1 and 2.

The main valve of the Meyer combination, Fig. 1, controls the lead, latest point of cut-off, the exhaust and the compression. The cut-off valve controls the point at which expansion shall begin. Comparing the two valves placed in mid-position of their travel, it will be seen that the edge *h* of the main valve will cut off the steam from the cylinder the same as the edge *K* of the common slide valve, Fig. 2. Also, the edge *J* of the one valve should act the same as the edge *J* of the other, providing the eccentrics are in the same relative positions. The same applies to the exhaust edges of each valve, *M* and *N*. The edge *h* will cut off steam from the port *b* independently of any action of the cut-off valves.

The longest distance at which steam is admitted is controlled by the main valve, and the work of the cut-off valve is to cut off steam at some earlier point in the stroke. This it does by the edge *G* of the cut-off passing over the edge *g* of the port *L*, and on the other stroke the edge *n* of the cut-off passing over the edge *n* of the port *K*.

Action of the Valves

For example, let the steam ports be 1 inch wide, both in the valve and the cylinder, and the main valve have a steam lap of ½ inch, which will give the main valve a cut-off at about nine-tenths of the stroke; and if the travel of the main valve be just sufficient to open fully the steam ports, the action of the main valve will not be unduly distorted by excessive lap or over travel. Assuming that the main steam valve and cut-off valve when in mid-position leave the edges *n* and *G* of the cut-off valve equidistant from the nearest edges of the main valve, i.e., one-half the width of the steam port, take these as average conditions and follow the movements as follows:

In Fig. 3 the crank (the travel being represented by the circle) is on its dead center at the crank end *B*, the cut-off eccentric in this example being set exactly opposite at *N* and the main eccentric at *M*, the main valve having no lead. It will be seen that the valves moving in the direction denoted by their arrows, while the crank starts in the direction shown by its arrow, the admission of

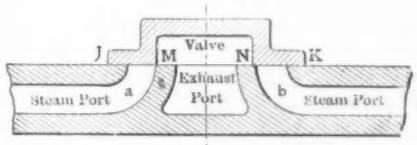


FIG. 2

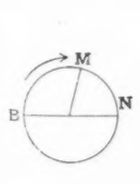
steam will occur through the ports *K* and *a* uninfluenced by the cut-off valve.

Fig. 4 shows the position of the valves, eccentrics and crank at the point of cut-off. The crank being at half stroke, the cut-off is performed by that valve independently of the action of the main valve.

Fig. 5 shows the position of the valves and eccentrics at the point of exhaust release on the same end from the port *a*, thus completing the action of the valves for one piston stroke.

Fig. 6 represents the same valves at one point in the processes shown in the previous figures. It is to illustrate the point that the effective area of admission of steam through the valve port *K* to the cylinder port *a* is governed by the action of the cut-off valve.

In this case, where the eccentrics are set as they are, the cut-off valve moves across the main valve port *K* before the latter valve has cut off, thus reducing the



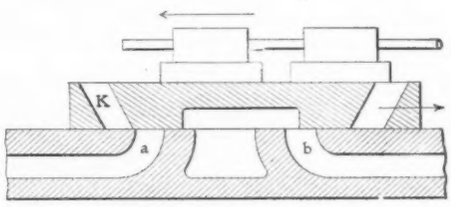
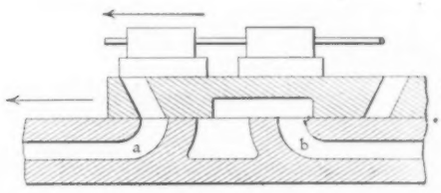


FIG. 3



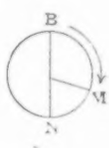


FIG. 4

effective port area. This condition must be considered to obtain the most effective use of the valves and a change in position of eccentrics to get the quickest action of the valves at the proper time is desirable.

Eccentrics are set as in the foregoing examples mostly on engines where a reversing of direction of rotation is wanted. But where the direction of travel is constant, a greater latitude of position of the

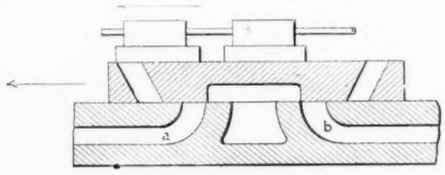


FIG. 5

eccentric is allowed and taken advantage of in greater or lesser degree by various builders. Assume that we still have valves with the same amount of lap, etc., as already shown, and study the effects of change of position of the eccentric by a few more illustrations.

In Fig. 7 the circle represents the travel of the crank and eccentrics. The crank is on dead center *B*, ready to start in the direction of the arrow; *O* is the throw line of the crank *N* of the cut-off eccentric and *M* of the main eccentric, *N* being 169 degrees ahead of the crank and 59 degrees ahead of the main eccentric, the angular advance of which is 20 degrees.

Fig. 8 represents the position of the valves and eccentrics at the point of cut-off. Bearing in mind the position of the eccentrics before illustrated, note the action of the same as shown in this figure. In passing the point *X* the eccentric moves the valve quickest on this stroke. It will then be seen that the cut-off valve has moved its fastest while near cut-off and the eccentric was passing this point; at the same time the main valve has not come to its best speed.

Then noting Fig. 9, which shows the relative positions at point of cut-off of the main valve, it is seen that the two valves have been traveling at nearly the same speed while making the cut-offs, with the

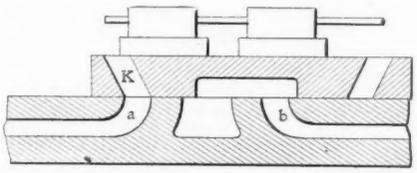


FIG. 6

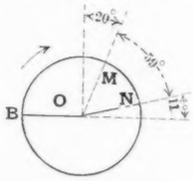


FIG. 7

sent the length of the eccentric rod, place one point at *M* and mark on the line *L—L* the arc *O;* next set one point at *X* and scribe the arc *P;* then with one point at *N,* scribe arc *Q*. The distance form *O* to *P* is the distance the valve traveled while the eccentric moved from *M* to *X,* and the distance from *P* to *Q* is the distance the valve traveled while the eccentric moved from *X* to *N*. The difference in valve travel while the eccentric traveled equal distances from *X* is shown by the arc *r*. This will illustrate why the cut-off valve must travel at least as fast as the main valve from the position shown in Fig 8 to that shown in Fig. 9.

Now comes the question of length of the eccentric rod. Assuming that in all the cases spoken of in this article an engine of 20-inch stroke were used, it will be found that in the case of setting the cut-off eccentric exactly opposite to the crank, the cut-off of steam occurred at 9 inches of both strokes, while in the case where the cut-off eccentric was set back 11 degrees in Fig. 7 the cut-off of steam is delayed to the thirteenth inch of one stroke and the sixteenth inch of the other, delaying the cut-off in both strokes and making it unequal. The additional reason for this latter condition is that in this case the cut-off eccentric has been set back and its rod made

steam port *L* is not yet closed. If, to close it, the cut-off eccentric be moved ahead, it will hasten the point of cut-off for the other stroke, and if we lengthen the rod to close *L*, it will *delay* the point of cut-off for the other stroke, and that would in this case cause a reopening.

The course to pursue is to change the position of the eccentric and the length of the rod as well, adjusting the two until the ports just escape reopening and the cut-off is equal. In this way the longest possible equalized points of cut-off are obtained.

Allowing that in every change of position of the cut-off eccentric a corresponding change of length of rod must be made: Referring back to Fig. 8, with eccentrics set as here (and the two valves moving together after the point of cut-off), the cut-off valve will not act on the forward stroke after the thirteenth inch of piston stroke, and since the main valve cuts off after the eighteenth inch of the stroke, therefore no cut-off can be effected between these points.

That this cannot be remedied by moving the cut-off eccentric may be shown as follows: Referring again to Fig. 8, if the cut-off eccentric were moved farther ahead, increasing its angle of 169 degrees, the cut-off would occur earlier, while if the angle were diminished, the cut-off would not be effected by the cut-off valve, because it would not fully cover the port.

This will be seen by reference to Fig. 12, in which the cut-off eccentric is moved closer to *X*, and as a result the cut-off is effected by the main valve. Finally if the cut-off eccentric were moved to the position *X*, Fig. 13, at the time the main eccentric stood at *M*, the cut-off valve would not effect the cut-off at all.

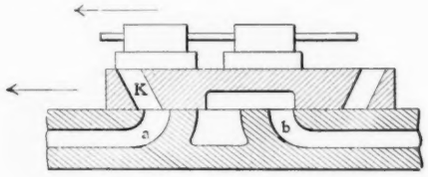


FIG. 8

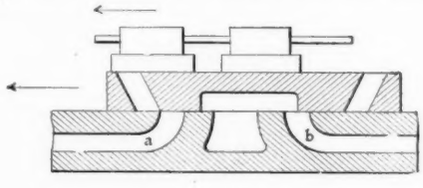


FIG. 9

cut-off valve going the fastest at the right point and slower than the main valve after the latter has closed the cylinder port, thus shutting out the possibility of reopening by the main valve's catching up and passing the cut-off before the main valve has cut off.

The main valve is traveling faster in Fig. 9 for the added reason that it travels faster while the eccentric is moving up to of such length as to get the longest point of cut-off without a reopening.

To equalize the points of cut-off it is necessary to move the cut-off eccentric back more and shorten the rod. In Fig. 11, for example, the positions of the valves are shown at the time the piston is on the thirteenth inch of its motion on the back stroke (this being the point of cut-off for the other stroke), and it is shown that the The cut-off eccentric is thus shown to be in position to cut off at the latest possible point, without reopening the port, as shown in Fig. 8.

Having limited the position of the cut-off eccentric in one direction, the next step is to find how far it can be set to cut off as early in the stroke as possible.

In the foregoing examples the cut-off was set at 180 degrees from the crank, or

at a lesser angle ahead of the crank, but it may be set at some angle behind the crank instead of ahead of it. It is understood that *ahead* means less than 180 degrees ahead of the crank in the direction of rotation, and *behind* the crank is less than 180 degrees in the direction opposite the direction of rotation from the crank.

In Fig. 14, for instance, it is set at 90 degrees behind the crank, the valve lap

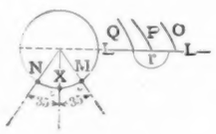


FIG. 10

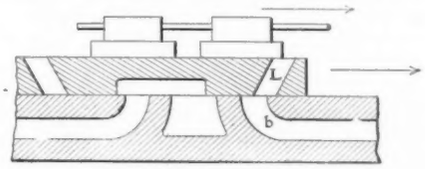


FIG. 11

and travel remaining the same as before, the crank *B* on "dead center," the main valve having no lead, and the port *a* being closed. In Fig. 15 are shown the positions at the point of cut-off, the crank having moved but 22 degrees.

On continuing the motion, the ports will arrive at the position shown in Fig. 16, from which it will be seen that the cut-off eccentric being at *N*, and the main eccentric at *M*, and the crank at *B*, the movements of the two valves would be in opposite directions. The cut-off valve will, therefore, keep the port *K* closed until the valves again change direction and approach the positions shown in Fig. 14; and it is clear that if the cut-off eccentric were set at less than 90 degrees behind the crank, the cut-off valve would first effect the cut-off, then lag behind and reopen the port *K*. In Fig. 17, for example, the cut-off eccentric has been advanced to 86 degrees behind the crank, and the cut-off valve after having cut off the steam lags behind and has begun its return stroke, allowing the ports to reopen and live steam to re-enter, as denoted by the arrow. Therefore, the least possible angle behind the crank for a cut-off valve is 90 degrees. These are the principal points to be understood and should be quickly grasped when ready to set these valves.

The ranges of angle of advance for both the main and cut-off valves above given are the maximum in either direction. In engineering practice, various builders vary the amount of advance and valve lap, but it is within this range. Some designers have less lap than herein given, say 25 per cent. of lap, and so adjust the position of the eccentrics that the cut-off will conform to the condition of the lap. In a multiported cut-off valve is exhibited the greatest degree of perfection, as it gives a maximum opening with a minimum lap.

It is not the writer's purpose to go too deeply into the design of this valve, only just sufficiently to set the valves with the hints that follow. Before going further, however, it is well to dwell on the principal points already brought out:

First, all classes of the Meyer cut-off valve are similar in action to the D slide valve, and easier to set for complicated conditions.

Second, Figs. 3, 4, 5 and 6 have to deal with single valves and eccentrics, with the cut-off set exactly opposite the crank. Such valve and eccentric arrangement is found mostly on marine engines, sometimes using a hand wheel. The use of the hand wheel on the valve stem gives the effect of more or less lap.

Third, where the direction of rotation is constant, and especially in stationary practice, a modification of the position of the cut-off eccentric is more desirable and is varied, as shown by Figs. 7, 8, 9, 10, 11, 12, 13, 14, 15, 16 and 17. Where the lap is 50 per cent. of the port, the following facts are to be remembered in locating the position of the eccentric: That to get a cut-off later than one-half stroke, the eccentric must be set less than 180 degrees *before* the crank down to 169 degrees before the crank (no less), as shown in Figs. 7, 8 and 9. To get a cut-off at earlier than one-half stroke, the eccentric must be set at less than 180 degrees behind the crank, down to 90 degrees behind the crank (no less), as shown in Figs. 11, 12, 13, 14, 15, 16 and 17.

Fourth, the multiported valves may change this so that in some cases, with small lap, an early cut-off may be secured with the eccentric less than 180 degrees ahead.

Fifth, in engines equipped with multiported valves, the length of rods, eccentric travel and valve lap are all attended to by the designer, so that the essential for the engineer in charge (after wear has set in and slipped eccentrics are a possibility) is to know the proper position of the eccentric and how to locate that point practically with all the gear mounted on the engine.

Sixth, with all cut-off valve gear the main-valve action is first looked into, and its action is to be considered the same as in any D slide valve. Then the cut-off valve is to be considered as the steam edges only.

To Set the Valves

On all engines, when preparing to set the valves, the first thing is to find and adjust all lost motion in the valve gear, then proceed to place the engine on the "dead center." Owing to the fact that the crosshead will remain stationary a short time before the crank pin passes the center, and remain so until the crank has passed the center a short distance, it is best to take greater care than to simply note the point where the crosshead rests at the end of the stroke, in order to find the "dead center." The writer believes the process illustrated in Fig. 18 is an accurate process.

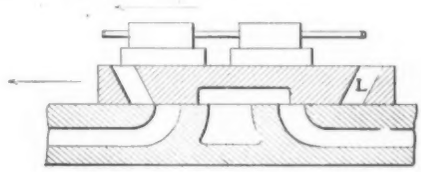


FIG. 12

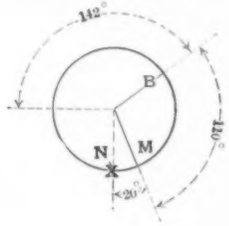


FIG. 13

In the figure, *J* and *K* represent two crosshead guides, *L* the crosshead, *M* the path of crank travel, *N* and *O* the center lines of the connecting-rod at different positions, and *P* the balance wheel. To start with, make a tram *S* out of any material convenient, preferably round steel. Turn the engine around until the crank is at the point *B* in any position where the crosshead is still moving with the other parts, near the end of the stroke. Then place mark *A* on the crosshead, running the mark up close to nearest or most convenient guide bar. (Where the crosshead is down in the frame of the engine, use a straight edge across the top of the holding-down bars.) Then place mark *C* on the guide bar opposite the mark *A* on the crosshead. Now, with the tram *S* (one point on the given mark on the floor op-

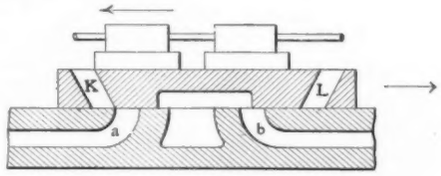


FIG. 14

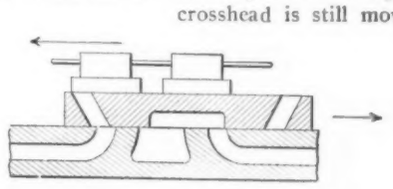


FIG. 15

posite the fly-wheel, and on a line with one edge of the rim) scribe the mark *D* on the face or side of the rim, making the arc come to the edge in either case.

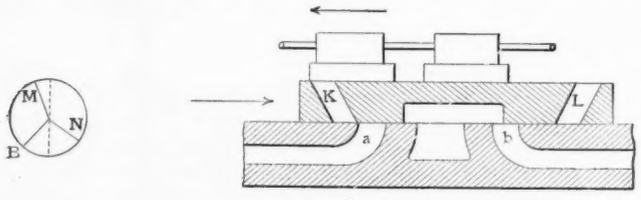
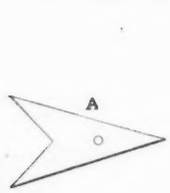


FIG. 16

Next turn the engine so that the crank passes the center and the mark *A* on the crosshead again comes to the mark *C* on the guide bar. Then take the tram, again resting it on the same point on the floor, and scribe the arc to the mark *F* on the same edge of the rim of the wheel. The crank will then be at *E*.

Now place prick-punch marks as near the edge as possible at the points *D* and *F*, and with a pair of dividers bisect the distance from *D* to *F* and make the mark *G*. Turn the engine back so that with the tram resting at the same point on the floor, the other point of the tram will touch *G*. The crank will then be on "dead center" at *I*, and the mark *A* on the crosshead will be opposite the mark *H* on the guide bar, which should be marked as the point where the crosshead reaches the end of the stroke.

Turn the engine to the opposite end of the stroke and repeat the foregoing moves, when both "dead centers" will



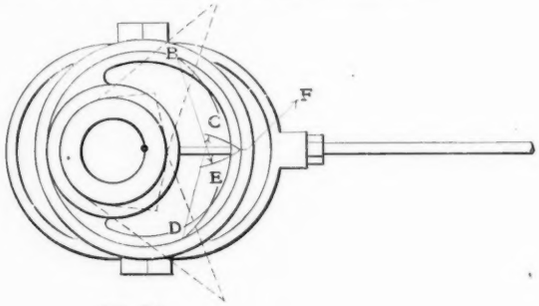


FIG. 19

have been found and marked. Now, with a marking chisel go over all the scribe marks and make them permanent, and where the center punch marks are on the rim of the wheel, the floor and the frame, it is a wise precaution to place marks like this ☐ around the center punch marks. It is also a good means of finding the marks in future.

The next move is to get the eccentric on "dead center." Where the eccentrics are fastened by set-screws, friction keys, or keys easily withdrawn, loosen up on one or the other, as the case may be, and turn the eccentric around the shaft while finding the centers. Where there is a fixed eccentric, as in governor eccentrics, the engine itself must be turned around to find the points desired.

Make a tram *A*, Fig. 19, out of a board or sheet steel and place at the point indicated a nail or other pointed iron or steel, just far enough out so that arcs *B C* and *D E* can be scribed, bringing the arcs down to the edge of the eccentric at the points *B* and *D*. Care should be exercised in using the tram, to have the end on the shaft or the boss of the eccentric the same distance away from the eccentric in both instances. Take a pair of dividers and from the points *B* and *D* scribe arcs so that they will exactly intersect on the eccentric edge at *F*. This will be the center line of the eccentric.

Make another tram *G*, Fig. 20, make a mark *H* with the center punch on the eccentric-rod, and with one leg of the tram on the point *H* scribe the arcs *J K* and *L M* on the eccentric strap coming to the points *J* and *L* on the edge of the strap. With the dividers scribe from points *J* and *L* arcs exactly intersecting each other at the port *N*.

With a marking chisel make permanent marks at the point *F* on the eccentric and the point *N* on the eccentric strap. Bring the eccentric around so that both points correspond and the eccentric will then be on one "dead center." From the points *J* and *L* scribe the arcs *O P* and *Q R*, ending at the points *O* and *Q* on the edge of the strap. From the points *O* and *Q* scribe arcs intersecting at the point *S* on the edge of the strap. With the chisel make a mark on the strap at *S*. Bring the eccentric around so that the point *F* corresponds with the point *S*, and the eccentric will be on the opposite end of the travel.

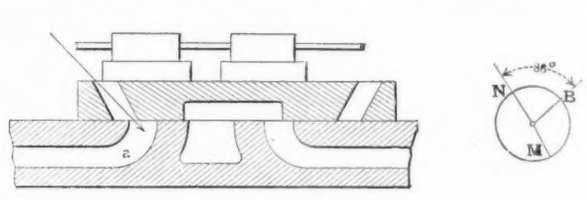


FIG 17

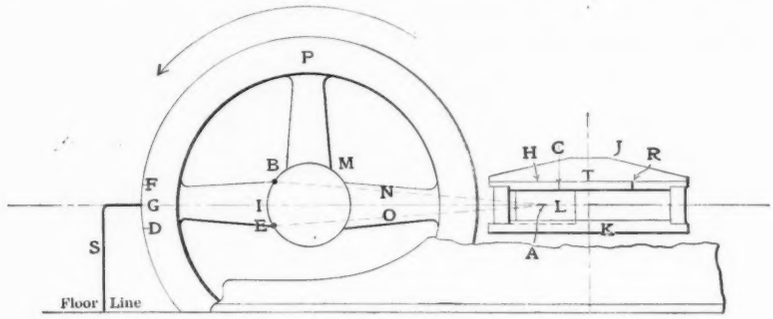


FIG. 18

Now assume an engine equipped with the Meyer cut-off, substantially as in the illustrations, i.e., a single main valve and single cut-off, each driven by one eccentric. Also assume that the builders have made all proportions, lap, width of ports and bridges correct. Take out the cut-off valve and start to adjust the action of the main valve, bearing in mind it should be set practically the same as any D slide valve.

Fig. 21 shows a steam chest with the valves removed. Take a scale and scriber and mark lines from the lower end of the ports *A* and *C* on the valve seat down to the bottom of the valve chest, and with an adjustable square resting on the surface of the valve-chest flange carry out these lines to the points *a*, *b*, *d* and *e* on the

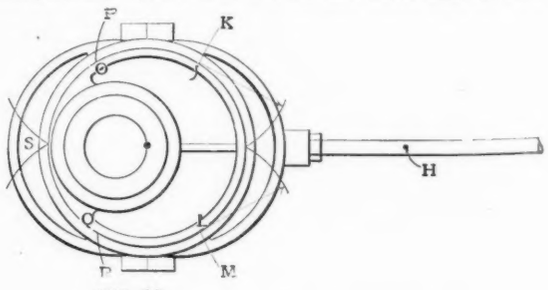


FIG. 20

valve-chest flange. Then bisect the distance between the inner edges of the steam ports, get a center line through the middle of the exhaust port *B* and carry it out in the same way to the point *c* on the steam-chest flange.

In Fig. 22 is shown the main valve in position, with the center line *A* corresponding with the center line *c* on the steam-chest flange. This shows the valve on the center of travel.

With the main valve in position and connected up, proceed as shown in Figs. 19 and 20, and place the eccentric on the "dead center" nearest the engine cylinder. With a scriber mark downward on the edge of the valve farthest from the eccentric, which will result in the line *m* on the valve seat (Fig. 21.) Move the eccentric to a point opposite the center and scribe downward along the same edge of the valve, which will give line *k* on the valve seat. With the dividers or scale locate and mark the line *l* on the valve seat, half way between the lines *k* and *m*. Then turn the eccentric around until the same edge of the valve comes to the line *l* The center line *A* on the valve should now correspond with the line *c* on the steam-chest flange, as shown in Fig. 22.

With the valve in mid-position, reach through its ports *a* and *b*, Fig. 22, with a scriber and mark along its steam edges on the valve faces the lines *i* and *j* (Fig. 21).

Upon removing the valve again it can be seen what the lap is, and whether it is even, by measuring from the line *i* to the nearest edge of the port *A*, and from the line *j* to the nearest edge of the port *C*.

When it is impossible to see the valve seat with the main valve in position, so as to scribe the lines *k*, *l* and *m* (Fig. 21) on the valve seat, proceed as follows:

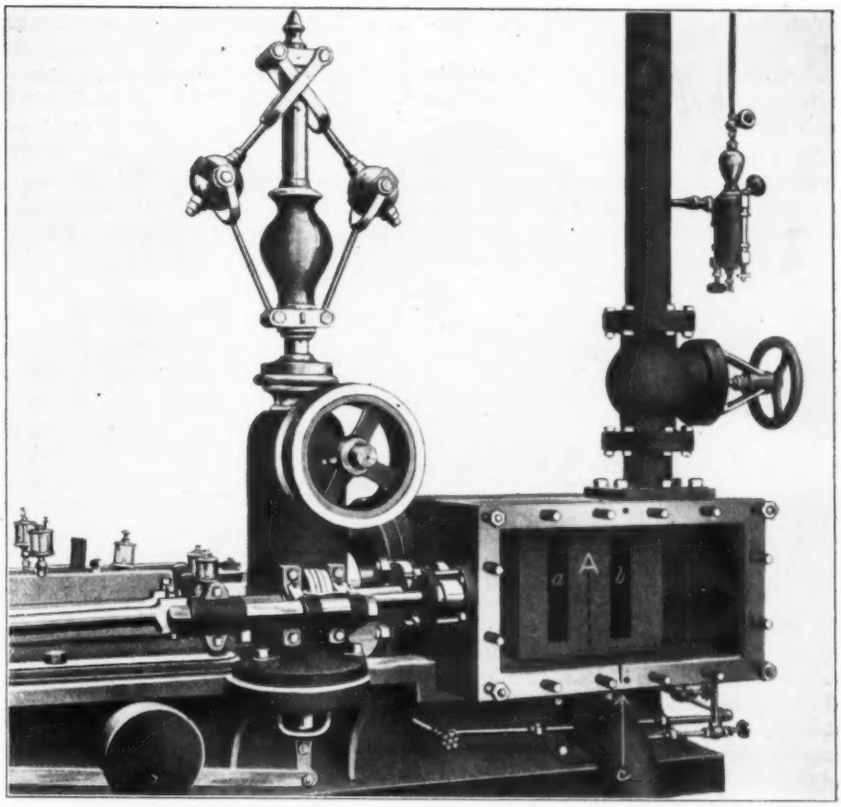


FIG. 22. SHOWING THE MAIN VALVE IN POSITION

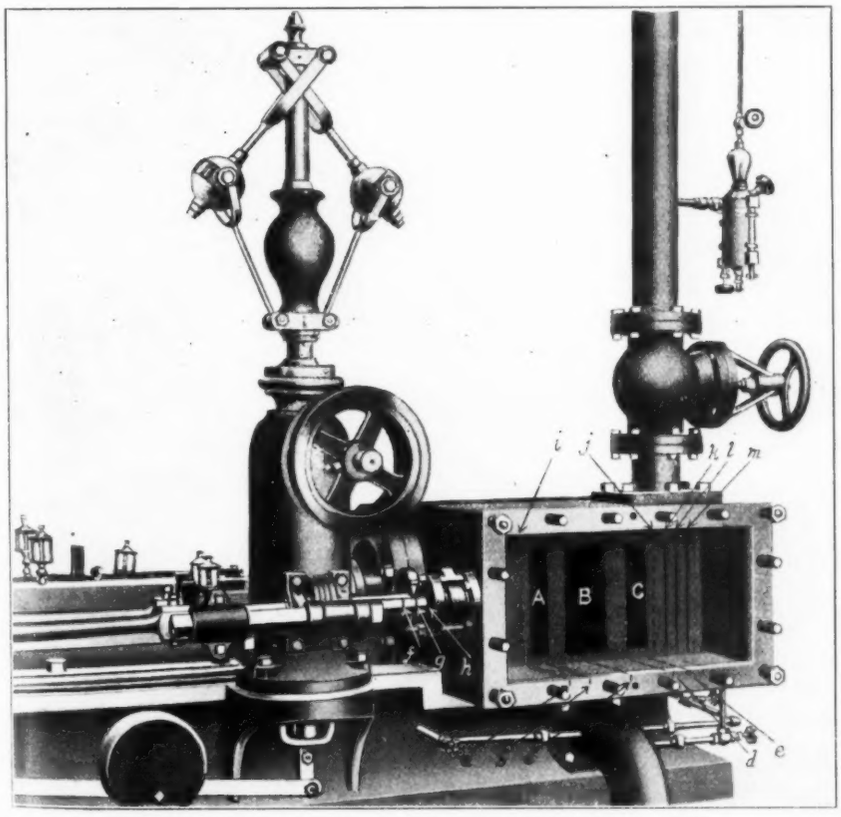


FIG. 21. STEAM CHEST WITH VALVES REMOVED

After marking the port lines and center on the valve-chest edge, as shown in Fig. 21, replace the valve and, placing the eccentric on the center nearest the cylinder, with the scriber mark a line *f* on the valve stem where it leaves the stuffing-box gland. Placing the eccentric on the opposite center, mark a line *h* on the valve stem at the same point at the edge of the gland. Find and mark the line *g* half way between *f* and *h* and turn the eccentric so that the line *g* is just entering the gland. The valve should be on the center of its travel and its center line *A*, Fig. 22, should correspond with the line *c*. If this latter process is followed, care should be taken to have the stuffing-box gland pulled up so that it will not move in or out with any movement of the stem.

In Fig. 22 the marks are shown on the cut-off valve stem, as it is easiest seen, but the same application applies to both stems.

Now place the engine on "dead center" and pull the eccentric around until the lead is given and then fix the eccentric in position. Pull the engine around to opposite "dead center," and observe the lead on that end. Make the lead the same on each end, or make the usual allowances for the crank end of a horizontal engine or the bottom end of a vertical engine, as determined by the builder. Place the cut-off valve in position connected up to valve stem. Use the same precautions on the cut-off eccentric as

with the main eccentric to get "dead center."

Then when the main valve is in mid-position place the cut-off valve in the same position, and observe if the cut-off edges are the same distance from the steam edges of the main valve. If they are, set the engine at one-half its stroke. To do this, divide the distances *H* and *R* on the guide *J*, Fig. 18 (which distances denote the end of the stroke), so as to get point *T*. Let the mark *A* correspond to *T*. Now pull the cut-off eccentric around until the cut-off valve edge is line and line with the cut-off edge of the main valve. Set the cut-off eccentric and pull the engine over to one-half stroke, going in the opposite direction. Observe if the cut-off is the same. If not, then the eccentric must be set ahead or back to even it up.

With both eccentrics set and secured in place, turn the engine around and observe that there is no reopening of the cut-off valve after it has cut off and before the main valve has cut off. Bear in mind the points brought out as to position of the eccentrics and lap of the valves in reference to cutting off before or after one-half stroke.

Where cut-off valves are operated by automatic governors all the rules here mentioned are to be observed when the governor weights are resting at the inner position of their travel with everything connected up. After the valve setting is all done, the governor spring or springs should be disconnected and the governor parts blocked out to the other extreme position. Then while the engine is turned a complete revolution, it should be observed that the cut-off valve covers the main valve ports at all points of the revolution. For finer adjustment of the valves recourse must be had to the indicator.

Single-eccentric Meyer Valve Gear

By H. V. Haight

The accompanying drawings ilustrate a Meyer valve gear in which both main and cut-off valves are driven from a single eccentric. This gear is used on the steam cylinders of straight-line air compressors.

Referring to Fig. 1, it will be seen that the main valve is driven by pin *C* on the eccentric strap, while the cut-off valves are driven by pin *D* on a rocker, which in turn is driven by pin *B* on the eccentric strap. The rocker is pivoted to the pillow block at *A*. This style of valve gear not only does away with one eccentric, one eccentric strap and one rocker, but it enables all the pins to be supported at both ends and placed so that the bearings are in direct line with the stresses, and there is no twisting action on the rocker. The cost sheets show this style of valve gear to be cheaper than the ordinary style using two eccentrics and two rockers.

It will be noted that the machine runs "under." In order to make it run "over" it would be necessary to drive the main valve from the rocker and the cut-off valves from the eccentric strap, which makes an awkward construction and does not give as good a valve diagram.

Laying Out the Gear

The method of laying out a valve gear of this kind would be about as follows:

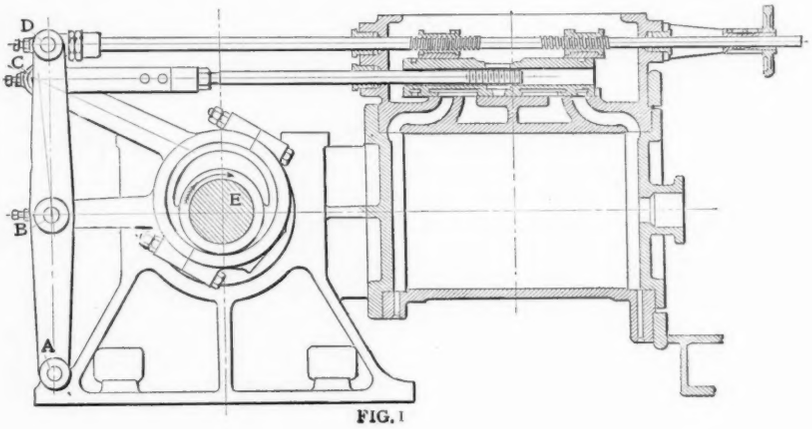


FIG. 1

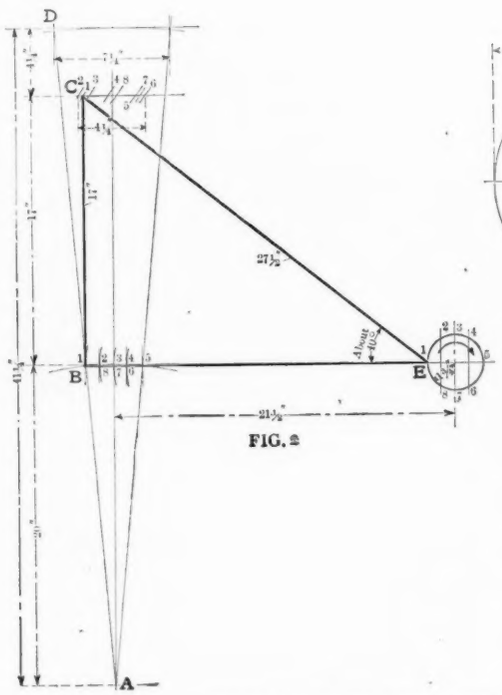


FIG. 2

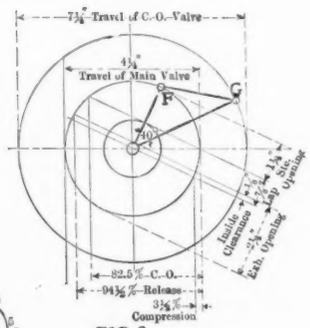


FIG. 3

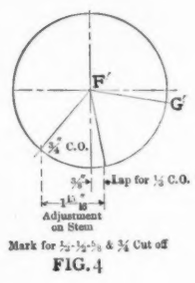


FIG. 4

SINGLE-ECCENTRIC MEYER VALVE GEAR

In laying out the cylinder, the length of the ports will be made about three-quarters the diameter of the cylinder, and the width will be made such as to give a velocity of 4000 to 5000 feet per minute. The steam opening will be made about three-quarters of the width of the port. The trial valve diagram for the main valve may then be laid out. For air compressors the valve has no lead and the compression is made small. The point of release will depend somewhat on the piston speed. The inner part of the valve diagram in Fig. 3 shows suitable percentages of cut-off, release and compression for a valve gear for this purpose. Having obtained the travel of the main valve, the next step would be to make the valve-gear layout, Fig. 2. An even figure would probably be chosen for the throw of the eccentric, which might bring the travel of the main valve slightly greater or less than that in the trial valve diagram. The final diagram may now be laid out. The angle between *B*—*E* and *C*—*E*, Fig. 2, gives the angular advance of the virtual cut-off eccentric, while the diameter of the outside circle in Fig. 3 will, of course, be made equal to the travel of the cut-off valves, as shown by Fig. 2.

Fig. 4 is then laid out by making *F'*, *G'* equal and parallel to *F*, *G*. The other positions for *F'*, *G'* are drawn for the minimum and maximum cut-off desired.—*American Machinist.*

Trials of Engineers in the Early Days—III

Interesting Results of Research in Rural England for Engine Types of the Time of Newcomen and Watt

EDITED BY ALONZO G. COLLINS

"I was one day returning from the town when I wondered at the engine going with great regularity, as formerly if the lad, being more intent on the games of his mates, who brought them near to tantalize him, than on working the engine, it would stand for the space of a stroke or two, until myself or the fireman, heaving a lump of coal at him, or fetching him a clout on the ear, he would be more nimble again; but now I had been harkening, for the space it may be of half a mile ('Ye gods!' ejaculated Rogers), and wondered greatly who might be at it, the beating was as regular as the surf on the beach; and, making all speed to get to it, I slippered around from the back side to take him unawares—and then I marveled greater yet, for the lad Potter was at a game of 'taws' with his mates.

"'Why are you idling here,' I cried at him, 'and who is working the engine?'

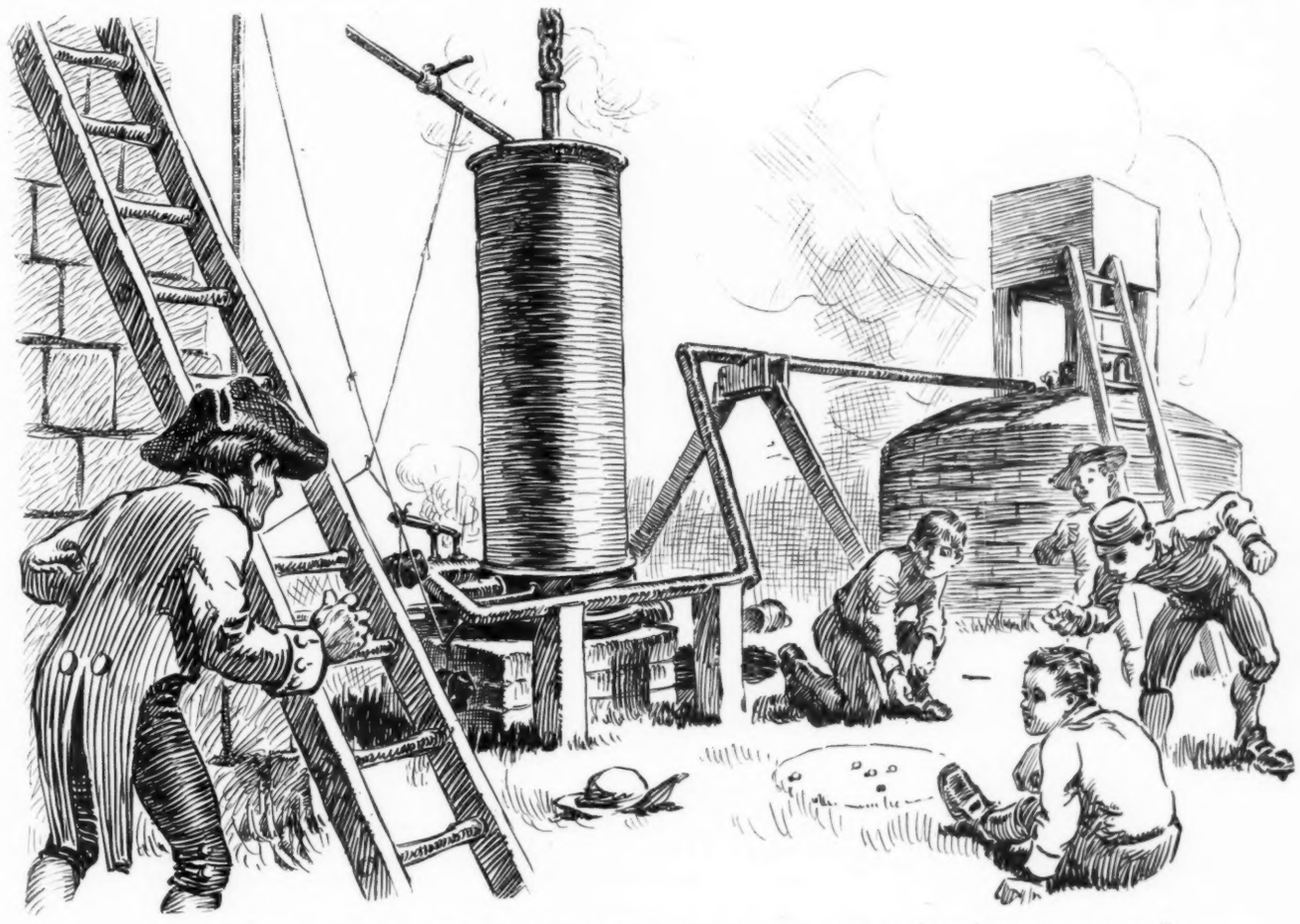
"'I just made a bit *scoggan*, zur, to make they cocks go by they beam,' said he, at which I turned to the engine and saw a tangle of cords and lumps of coal, going this way and that, from the great lever, and no soul near it; and the thought that came to me was, 'Out of the mouths of babes and sucklings.'

"Humphrey Potter had saved his strength to some purpose, albeit it was but a crude affair and would not serve its purpose for long; but I presently discovered a way for the betterment of it by rods and lumps of lead, and the going of the engine was made so much better that now it went twelve strokes in a minute.

"But I still had but little idle time; the pump leathers gave much trouble, being soon spoiled; but I had always some by me, and so the engine was always speedily going again, albeit there was little pleasure in standing up to my middle in water while getting at it.

"There was also much trouble with sticks and such like coming in the pump through the sucking pipe to the clacks and letting the water run back; but I contrived a square frame of wood, with holes round its bottom in the water, to inclose the lower end of the sucking pipe, to keep away the obstructions. And the augmented speed at which the engine went enabled me to sooner clear the pit."

"I was now considering much how the engine might be made speedier, as it might be if the steam might be sooner condensed, but could discover no way by which this might be done; and then a surprising thing happened, as I saw the engine go several strokes, and very quick, and I wondered much why this should be, but observing the water issuing from the



"AND THEN I MARVELED GREATER YET, FOR THE LAD POTTER WAS AT A GAME OF 'TAWS' WITH HIS MATES"

snifting valve in greater quantity, I searched in the mouth of the cylinder and saw the water did not lie long on the piston, and discovered where a hole had gone through the bottom of the piston, through which the water went into the cylinder; and I reckoned that the steam being more intimate with the water, was the sooner condensed; and I considered how I might contrive to inject water into the cylinder, in such manner that it might be regulated.

"This I compassed by fashioning a pipe in the cylinder bottom, with a cock to regulate the water, and I stopped the hole in the piston, with a screw-bolt and a grummet fixed firm. And this so increased the efficacy of the engine, that it would now go sixteen strokes in a minute; save that the boiler could not then boil sufficient steam for the engine, no matter how fierce the fire.

"But I still must put water on top the piston, to keep air from going in, as I had not been so far able to pack the piston so it would not soon let air in.

"There was now a great quantity of water, scalding hot, issuing from the snifting valve; and I considered greatly how I might contrive to get it to replenish the boiler, as I reckoned it would be to great advantage, and to test the matter, I contrived a cistern in the earth, where the hot water might go, and set men to pull up the hot water with buckets and ropes, and pour them in the little cistern above the boiler, meanwhile letting no cold water in; and this for days together, and this so augmented the boiling as the engine would go about sixteen strokes in a minute, and no more coals were consumed than before.

"I had now to contrive a pump by which to lift the hot water to the feed cistern, but I must need first to know how great a quantity the boiler must have for replenishing the boiling for one stroke of the engine. This I learned by many experiments with measuring the water supplied to the feed cistern to be about three and a half gallons in a minute, as fast as it consumed in boiling, and the engine going sixteen strokes in a minute, which I reckoned about one-fourth of a gallon for one stroke; but to be certain I made the pump to supply a half gallon to the stroke; the pump to derive its water from a hot-water cistern in the earth, and to lift it by a lead pipe to the feed cistern above the boiler; and so much of the water as was not needed to replenish the boiler, should waste away in an overflow pipe.

"The feed-pump I made go by a rod from the great lever, but I was greatly surprised that I could get no water from it, and this did much humiliate me, as I had been of late much puffed up; but I made diligent search to know what could be the cause, and could find nought amiss with the clacks or bucket, nor yet the sucking pipe ('How many engineers have pulled a pump to pieces since then,' thought Reynolds), but discovered when the water was cold the pump could lift it, but when the water was hot it could not; and at this I reckoned the vapor from the hot water went through the clacks, so there was no space for the water, and my device was like to come to naught.

"I then considered if I might elevate the hot-water cistern, as so the water might come to the pump from the bottom thereof and the vapor pass away from the top, and, this being done, the pump answered my service. As for the reason of this, I gave much further study and reckoned that when the pump bucket ascended, the vapor, being much rarefied, filled the pump's bore and no water might come in, and when the bucket descended the vapor was pressed down and, being of less weight than the water, had no force to lift the upper clack and so free itself, but remained in the pump.

"I was now having great trouble with the great lever; and again did my ignorance contrive to drag down my pride, for the lever, being of so many different parts, would have been better for more consideration than I gave to it, and I must now contrive to make it stanch, as it bade fair to be utterly spoiled soon. I got new bolts and set one between each of those that were in before, but to very little benefit; and while considering how I might better it, I observed that where the timbers were together they rubbed one against the other, so to wear away as to be presently slack, although but a short time before the bolts had been screwed firm.

"I now reckoned if this rubbing might be prevented, it would be to great advantage, and if oak blocks might be set in the joint, half in each one, it might suffice, and I bargained with a clever wheelwright, who should take advantage of the times when the engine was not going, by reason of packing the piston, or other difficulties, or when there was no water in the pit (for with the superior speed at which the engine now went, the pit was at times clear of water for a space), and work mortises through the lever, half above and half below each joint, in which with heavy hammers we drove oak lugs and to such advantage that the beam was as stanch as might be.

"I had now, as I thought, got the engine so it was stanch and able to go a long time before being spoiled, and it was a matter of great satisfaction to me that the engine did discharge as much water as did before employ more than fifty horses at a cost not less than £900 a year; whereas the fire engine bid fair to never, in coals, attendance and repairs, cost more than £150 a year, and the license, £80 a year.

"I had of late been much concerned as how to reckon the proper proportions of the fire engine, as we had luckily by accident hit on what we sought for in this one. The depth of the pit I found 300 feet, and by my experiment with the feed cistern, the weight of the water on one inch would be 150 pounds; the pump's bore was five and three-quarters inches. I had no way of reckoning how many inches were in the circle of the bore, and I scribed on a smooth plank, with the smith's compass, a circle of five and three-quarters inches in diameter, and measured squares in it, one inch each way; the part squares next the boundary I measured in squares one-quarter inch each way, and the number of squares in it, after reckoning the little squares, I reckoned to be a little more than twenty-six, but to be certain, said twenty-seven inches in my reckoning, as this would be more, than less; so I reckoned the whole weight of the water in the pump and force pipe to be 4050 pounds.

"And now I was much concerned by the boiler letting out steam from the flanch joint, which presently became so troublesome that I was fain to break down the brick to get at it, and found where the putty had gone, leaving a hole for the steam to come out; but I compassed to stop it with oakum driven in where the putty had been; but at that it broke again at another place, and I must drive in more oakum, until I had at last encompassed the circumference of the flanch with cakum, and with the fire and smoke was a most fearsome task; but, having accomplished it, I was again at ease to get at my reckoning.

"I numbered the inches in the steam piston in the like manner as the pump bucket, and reckoned it closely 615 inches; on which the atmosphere pressed 15 pounds on each inch; and the whole force of the air I reckoned 9225 pounds. At this I was much surprised, it being so much greater than the weight of the water, and considered for a long time where I might be at fault, and then saw I had missed the weight of the pump-rod and bucket, and this I was ignorant of; but when the engine was stopped for a space I removed a piece of the rod, and the weight of it was closely 84 pounds, and I reckoned the nineteen pieces to be 1596 pounds, and the weight of the bucket as set out on the invoice 404 pounds, and the whole weight 2000 pounds and, with the water, 6050 pounds.

"But this was still much less, about one-third, than the force on the piston, and I wondered greatly how this must be; when I considered that mayhap I had not a good vacuum under the piston, and that some force was needful to cause the great lever to move up and down and overcome the momentum of the water in the pump, and other accidents; so I came to reckon that the force of the air on the piston must be in proportion about one-third greater than the weight of the water in the pump and force pipe, and the weight of the pump-rod and bucket; but I lamented much that in my ignorance I knew

no way to reckon the number of inches surface on any certain piston.

"I had now to make stanch the joints in the steam pipe between the cylinder and boiler, which by reason of the jumping of the cylinder were like to be spoiled, and I now saw where again I had been at fault: whereas the putty, after being a long time used, being hard and of a brittle nature, would not accept the jumping of the cylinder, but, becoming powder, left holes, where the steam came out, and I cast about for some material that would not be so soon spoiled, and in my perplexity I put leather in one, cork in another, and pasteboard in yet another; but all were soon spoiled, until I came by way of dipping pasteboard in linseed oil, and this was not spoiled for a long time, and again I might be at my reckoning, wondering meanwhile if I should presently get to the end of my learning.

"I must now know what quantity of steam a cubical inch of water produced. I found by many experiments that when the force of the steam was just one pound avoirdupois on a square inch, it was sufficient to work the engine and that about three and a half gallons in a minute would feed the boiler as fast as it consumed in boiling, and steam for the cylinder, sixteen strokes in a minute. The cylinder contained 222 gallons of steam every stroke, in a minute equal to 3552 gallons of steam, so three and a half gallons of water produced 3552 gallons of steam, and, 277.27 cubical inches in one gallon, one cubical inch of water produced 3.6 gallons of steam. Hence, it appears, one cubical inch of water by boiling till its force is capable of overcoming one-fifteenth of the force of the atmosphere will produce 1008 cubical inches of steam. But consideration here showed me that again I was at fault, as a part of the steam was condensed in the cold cylinder, before it was filled with steam and had become hot, and that the quantity of steam produced from one cubical inch of water must be greater than I reckoned; and I saw no way by which I might reckon this with any degree of exactitude; but to accomplish the task I had set myself, which was to reckon how great a boiler might be needful for any certain fire engine, I must know this.

"And now I was come to grief with the joint between the cylinder and the bottom which I could get stanch but for a short space of time, and consideration showed me that the lead was at fault, in that being hot one instant and the next cold; when it was hot, being soft, it was presently pressed thinner, and then becoming cold was slack, and I considered many ways by which to rectify this; pasteboard I could not contrive, for the great irregularities in the flanches, being in some places more than half an inch wide; but remembering how at the iron works some lathe turnings which were in a heap where the men relieved themselves did presently become hard, I sent and got some fresh turnings, wet them with urine and beat them into the joint with blunt chisels and hammers. till it was hard and gave out a ringing sound, as if I had beat the natural iron; and I had never, no more trouble with this joint, and once more I was at ease for a space of time, and might get at my contriving.

"I now considered how if one cubical inch of water should produce some certain quantity of steam at a force of one pound, that same quantity of steam if condensed and made cold must show again one cubical inch of water, and my task was now to contrive a vessel, by which I might exactly measure this; for the proportion of steam to water I reckoned to be so great that a small vessel, such as my means might contrive, would leave such a little residue of water that I would have no means to measure so minute a quantity.

"There was, however, at the iron works a cylinder that, having been inadequate for the pit it was destined for, had been exchanged for a bigger one; and I bargained with Mr. Darby—I to give my time gratis, and he to contrive a top for the cylinder, and otherwise prepare it to suit my purpose, and pay for the carriage of it to the pit, where I, as opportunity offered, should make such experiments, as would determine the matter, for our joint benefits; my intentions being, on some holiday when the pit men were idle, to contrive a steam pipe from the boiler to the cylinder set near by, and in the cylinder bottom to set a cock by which to let out the water.

"This being done, I opened the cock in the steam pipe, and the cock in the bottom, a little, and let steam in the cylinder, having care to regulate the cock and the fire, as that the force of the steam should not be abated less than one pound, nor more augmented; and when the steam issued clear from the cock in the bottom and the cylinder was hot, I shut them both. I had before measured the cylinder with a brewer's gallon, of which it contained one hundred and thirteen, which is nine hundred and four pints. When the cylinder was quite cold, I let out the water into a measure, and had a little more than half a pint; but I could get no great degree of exactness, as I could not contrive to collect all the water. I removed the top, and pushed down the drops which like dew clung to the sides of the cylinder, and so collected more, until I judged I had closely two-thirds of a pint. This I tried many times, always about the same, so it appeared to me that one cubical inch of water produced 1500 cubical inches of steam, with a force of one pound on a square inch, which was about one-half more than I had before; so that of the boiling, it appeared that about one-half was wasted.

"I must now determine what quantity of water might be boiled with a bushel of coals, and as I knew that three and a half gallons were boiled in a minute, which means two hundred and ten gallons in an hour, I had but to measure the coals for an hour to know it; and this, after a great number of experiments, was closely four bushels in an hour, which appears fifty-two and a half gallons to a bushel. To know the size of boiler needful, must be to know how much of it is to the fire, as no other part has any boiling effect, and to know this I must contrive how to measure this boiler. As it appeared there must be some certain quantity boiled on a square inch, or, on a square foot, or a square yard, and as I considered the matter, it presently offered to me that one square inch might be too minute, and a square yard too big a quantity to reckon conveniently, and I made two sheets of pasteboard, exact one foot each way, and going in the fireplace when there was no fire there, I laid one sheet of the pasteboard smooth against the boiler at the top of the fireplace, and scribed around it; then setting the other pasteboard against the first I scribed around the second one; then setting the first one against the second, on its other border, I scribed around it, and so on, one after the other until I had measured it all, and numbering each place I found 168 of them; so it appeared one and a quarter gallons was boiled in an hour on every square foot of fire surface.

"But no sweep was ever blacker than I when I came out.

"The fireplace where the grate was was four feet in width and five feet in depth, equal to twenty square feet, by which it appeared two-tenths of a bushel was consumed on a square foot in an hour."

"Gee," said Reynolds, "I reckon some of these young M. E.'s wouldn't do a thing to that evaporation test."

"The quantity of water lifted from the pit was closely 7900 gallons in an hour, although the pump had been reckoned to lift more than 8000 gallons in an hour; but by measuring the pump's discharge by the casks from the tub-gin, the mean was as first stated. By this I made the conclusion that a part of the water was lost past the bucket and the clacks, in the proportion of about one-twentieth part; so that it appears that when you have determined the quantity of water lifted from the pit by the tub-gin, the pump for such quantity must be reckoned for a quantity one-twentieth greater.

"I also reckoned that 7900 gallons could be lifted three hundred feet in one hour by the boiling of two hundred and ten gallons of water to a force of one pound of steam, and the consuming of four bushels of coals; and that if the lift was greater or less, so would the water boiled and coals burned be greater or less, in that proportion; or, if the gallons of water lifted three hundred feet were greater or lesser, so would the coals be greater or less.

"By experiment I found that from the

snifting valve there came out closely to three gallons each stroke, and I was much amazed how that steam which is made from about two pints of water should heat more than two and one-half gallons of cold water, to have it come out scalding hot, which it does.

"By this time I was come to know that if I was to be expert in this business, I must compass some way to get a better knowledge of how to reckon figures than I had."

Whether he succeeded in his laudable ambition, Reynolds was not to know, for the narrative came to an end, and further search was fruitless.

Catechism of Electricity—XXIII

PORTABLE DIRECT-CURRENT VOLTMETERS

478. *Are all voltmeters portable?*

No; voltmeters, or instruments for measuring the electrical pressure or voltage of a circuit, or the drop in pressure across a resistance, are not always portable; neither do all of them operate upon the same principle. The term "portable" signifies that the instrument is constructed so that it can easily be carried from place to place, in contradistinction to "station" voltmeters, which are constructed to be used permanently in one place, as on a switchboard in a station.

479. *Show a typical form of portable direct-current voltmeter.*

Fig. 106 shows such an instrument, removed from its case. This voltmeter is designed to measure direct-current pressures up to 150 volts. Two scales *m* and *n* are provided, the former graduated with practically equal subdivisions from 0 to 150 volts and the latter from 0 to 15 volts. Pressures up to 15 volts can be read on the low scale with greater accuracy than if read on the high scale, for the reason that the unit divisions are spaced farther apart. Pressures above 15 volts must, of course, be read on the high scale. In order that either scale may be used at will, three binding posts *a*, *b* and *d* are provided. When the binding posts *a* and *d* are used to connect the instrument in circuit, the low-reading scale will be in use; the binding posts *b* and *d* correspond to the high-reading scale. The circuit through the voltmeter is not closed, and consequently there is no deflection of the pointer *p* over the scale, until the button or key *k* is pressed down. The instrument is direct-reading; that is, the scale markings represent the exact values being measured.

480. *Show the working parts of this voltmeter and describe the principle upon which they operate.*

The working parts of the complete instrument shown in Fig. 106 are represented in Fig. 107. In a two-scale voltmeter there are two separate resistance coils *a'* and *b'*, Fig. 107, one for each scale. Each of these coils is wound with the same size wire, non-inductively; that is, the wire of each coil is wound back on itself as in the case of the resistance coils in the Wheatstone bridge. The wire on coil *b'*, which is used for the 150-volt scale, is nearly ten times the length of that on coil *a'* for the 15-volt scale. One terminal from each coil is connected to the binding posts *a* and *b* respectively, and the remaining two terminals are joined together and electrically connected to *c*, a fine copper-wire coil which is free to turn about a vertical axis. The other end of the coil *c* is electrically connected to binding post *d* through the key *k*. Within the coil *c* is permanently mounted a cylindrical piece of soft iron *s*, which acts as a core, thereby strengthening the magnetic circuit of the permanent horseshoe magnet *N S*. The coil *c* is wound upon a light frame of copper which serves both as a support for the wire and as a damper to make the instrument dead beat.

The pointer *p*, attached to the top of coil *c*, is attached to the coil. When no

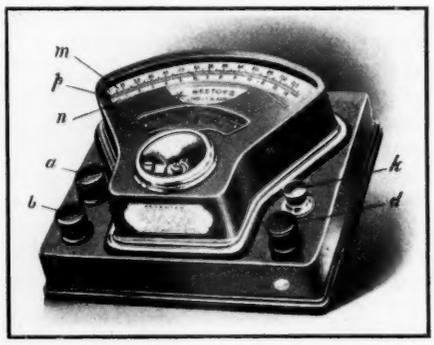


FIG. 106. A PORTABLE DIRECT-CURRENT VOLTMETER

current is passing through the coil, the pointer is over the zero mark on the scale. When current passes through the coil in the proper direction, a north pole is produced at the farther side of the coil, and a south pole at the nearer side. The repulsion between the like poles of the coil and permanent magnet, and the attraction of the unlike poles, cause the coil to turn so that the pointer is carried over the scale. The position of the pointer thus depends upon the magnetic effect produced by the coil, and this in turn is proportional to the strength of the current passing through the coil; as the current is proportional to the voltage applied to the binding posts of the meter, the internal resistance of the meter being constant, the position of the pointer is proportional to the voltage applied.

Two flat spiral springs *r* and *r'*, one fastened to the top, and the other to the bottom of coil *c*, exert a force in opposition to that produced by the current in the coil; consequently, when no current flows through the instrument the force of the springs brings the pointer back to the zero point, which, in this class of instruments, is at the beginning of the scale. These springs also serve to convey the current into and out of the movable coil. The moving parts are supported in sapphire jewels, and the pivots are of hardened steel, ground and highly polished, in order to minimize friction and obtain extreme sensitiveness.

481. *What would happen if current were passed through the meter in the wrong direction?*

A south pole would then be formed at the farther side of the coil *c*, and a north pole at the nearer side. The attraction of these poles by the poles of the permanent magnet *N S* would cause the coil to move in the opposite direction to that previously stated, and the pointer would be forced off the scale, at the left.

482. *Does not the movement of the pointer therefore indicate whether the voltmeter is correctly connected to the circuit?*

It does. The key *k*, however, should be but lightly tapped for this test, as otherwise the pointer is liable to be bent by being forced against the side of the case or against the insulated stop provided in some meters. If the deflection is in the wrong direction, the connecting leads should at once be reversed.

On the meter, Fig. 106, directly below the right-hand binding post is shown a white disk, and if the positive side of the circuit to be tested is known it can be connected to the binding post thus indicated, with the certainity that the pointer will be deflected in the proper direction.

483. *How is the proper resistance for the coils a' and b', Fig. 107, determined?*

Each of these coils has a resistance depending upon the full-scale reading which it governs; usually, the coils are wound to make the total resistance of the moving and stationary coils about 100 ohms per volt. The coil *a'* and the moving coil *c* therefore have a combined resistance of about $15 \times 100 = 1500$ ohms, and the coils *b'* and *c* a resistance of $150 \times 100 = 15{,}000$ ohms.

484. *What are the usual measuring capacities of portable direct-current voltmeters in common use?*

Portable direct-current voltmeters vary from 0.02 volt up to 750 volts capacity, the smaller ranges being used for the measurement of very low electromotive forces, such as those developed by difference in temperature of two metals and by battery cells or for measuring the voltage across very low resistances, such as armatures of dynamos, heavy cables, etc. The voltmeters in most common use are of 5, 15, 75, 150, 300, 500, and 750 volts range.

485. *How are direct-current pressures in excess of 750 volts measured?*

Direct-current pressures seldom exceed 750 volts, but it is practicable to extend the measuring range of a voltmeter many times by the use of a device called a mul-

tiplier. The multiplier is merely a special resistance connected in series with the voltmeter and is often used with low-reading voltmeters to extend their field of usefulness.

486. *Illustrate a multiplier and its principle of operation.*

Fig. 108 shows a multiplier. The perforated cylindrical case contains a resistance connected to the binding posts

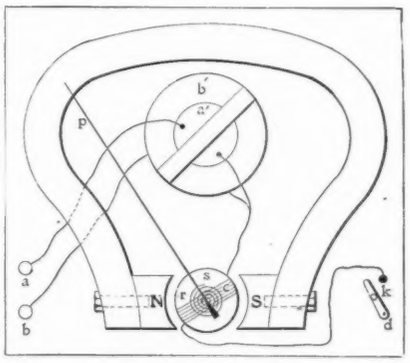


FIG. 107. CONSTRUCTION AND CONNECTIONS OF FIG. 106

m and n, the value of which is some multiple of the resistance of the voltmeter to be used with it; the resistance of the voltmeter circuit is thereby increased a certain whole number of times, and consequently the deflection on the voltmeter scale is decreased in exact proportion. The readings on the voltmeter connected in series with the multiplier have, therefore, to be multiplied by the proportion in

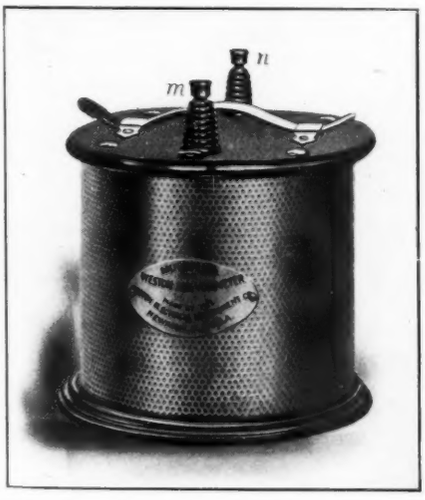


FIG. 108. MULTIPLIER USED WITH PORTABLE VOLTMETERS FOR INCREASING THEIR RANGE

which the resistance of the voltmeter circuit has been increased. The perforated case is used to afford ventilation for the wire within so that its resistance will not be affected by overheating.

487. *What is the voltage of a circuit in which a 150-scale voltmeter having a resistance of 14,812 ohms indicates 115.5 volts when connected with a multiplier of 44,436 ohms resistance?*

According to the answer of Question 486, the multiplying factor would be

$$\frac{14{,}812 + 44{,}436}{14{,}812} = 4.$$

The voltage of the circuit would therefore be $4 \times 115.5 = 462$ volts.

488. *Are any temperature corrections necessary in the readings obtained on the direct-current portable voltmeter by reason of the change in its internal resistance at different temperatures?*

The temperature correction in the meter illustrated is not over one-quarter of one per cent. between 35 degrees and 105 degrees Fahrenheit, and the instrument readily gives results within one-fifth of one per cent., if carefully used; for commercial work, therefore, the temperature corrections are negligible.

489. *How should a voltmeter be connected in circuit?*

It should be connected across that part of the circuit where the voltage is desired, by small flexible wire leads. If, for example, the voltage of a generator G, Fig. 109, be required, a voltmeter V must be connected to its terminals as shown. If the drop in pressure across a resistance R be desired, a voltmeter V' must be connected to the ends of the resistance. The high internal resistance of the voltmeter permits the passage through it of only a very small current; consequently, but a small amount of energy is required to operate it.

490. *What precautions should be observed in connecting up a voltmeter?*

A voltmeter should not be connected to a circuit unless one has a knowledge of the character of the circuit and an idea of the approximate voltage. If a voltmeter capable of measuring only 150 volts be connected across a circuit giving 300 volts, and the key be pressed, the resistance coil of the voltmeter, or possibly its fine wire coil, would be burned out. That is, the wire would not carry the larger current that would flow through the meter without heating to such an extent as to set fire to its insulation, or at least char it, rendering it unfit for further use.

Voltmeters, and in fact any measuring instrument depending upon the magnetic effect of a current, should not be used in close proximity to dynamos or conductors carrying large currents, because the readings on these instruments are affected by stray magnetic fields, and the steel magnets used in the meters may become permanently changed.

When measuring instruments of any kind are placed upon tables or benches and connected in circuit by means of loose wires or cables, these wires or cables should be securely clamped or fastened to the tables or benches. The instruments will not then be liable to injury if the wires are accidentally caught and pulled.

When carrying instruments or setting them down, they should not be handled roughly. Such treatment dulls the pivots and frequently breaks the jewels.

Care should be taken to prevent either of the connecting leads from touching the cover when a voltmeter is in use. The pointers in these instruments are often permanently connected with one side of the circuit and touch the case at the end of their swing. If a lead from the other

FIG. 109. HOW A VOLTMETER SHOULD BE CONNECTED IN CIRCUIT

side of the circuit also touches the case, a short-circuit will be caused.

491. *What precautions should be observed in taking readings on a voltmeter?*

In reading deflections of the pointer on the scales of all instruments, it is generally possible to read to tenths of a division with a fair degree of accuracy. In case a mirror is provided beneath the pointer, advantage should be taken of this feature by sighting the needle so that it covers its reflection in the mirror.

A slight jarring or tapping on the case of the instrument will overcome the friction of the pivot and aid the pointer in coming to rest. A separate key is sometimes used to stop the vibrations of the pointer, especially in the case of alternating-current instruments, as will be explained later.

If the pointer be bent so as to lie more than one-fifth of a division away from the zero mark when no current is flowing, the case of the meter should be unscrewed, taken off, and the end of the pointer bent back to the zero mark; this is best accomplished by means of a small pair of pliers.

Alcohol from Cactus Plants

G. A. Burns, a dentist of Los Angeles, suggested in a letter to the *Scientific American* that cactus is a cheap source of alcohol. Knowing that the natives of New Mexico make an intoxicating liquor from cactus, Mr. Burns experimented and, in a crude way, distilled one gallon of alcohol from five pounds of cactus. Cactus plants, he says, can be cut down, ground up, and put through the process of fermentation and distillation like corn, wheat, or barley. He adds that Nevada cactus grows from two to five feet high very quickly, and two months after being cut down it would grow up from the stubble and be ready to again produce industrial alcohol. Since in the West cactus grows abundantly and without irrigation or other attention, this may be one of the cheapest sources of industrial alcohol likely to be found in this country.

The Hydraulic Elevator—VI

Good Features of Magnetic Valve Control; Obstacles That Prevent Its General Use. Description of Control Systems

BY WILLIAM BAXTER, JR.

The running-rope and standing-rope arrangements for moving the pilot valves of hydraulic elevators enable the operator to control the movement of the car perfectly, i.e., to cause it to run at any desired speed, from maximum to a barely perceptible motion, and to start and stop as slowly or rapidly as he may desire; but, structurally considered, these arrangements are not entirely free from objection. Comparing them with the simple hand rope, it can be seen that they have several additional parts, and this alone increases the first cost and makes them more expensive to maintain. In practice, however, their operation has proved entirely satisfactory, with the single exception of their liability to get out of adjustment through the stretching of the ropes. When in proper adjustment, the car lever will stand vertical when the valve is closed. If when the ropes stretch they both lengthen out the same amount, the adjustment will not be disturbed, and the fact that the ropes have stretched will not be noticed, as the tension weight will take up the slack; but if one rope lengthens out more than the other, then the car lever will not stand vertical when the valve is closed, but will incline to one side or the other. If the difference in the stretch of the ropes is not great, the "stop" position of the lever will vary so little as to make no difference in the operation of the car; but in some cases the lever will stand several inches from the central position when the car is stopped, and then it becomes necessary to readjust the ropes. This is accomplished by taking up the rope that has stretched the more, until the car lever is brought to the central position when the valve is closed. When the ropes are new they may require frequent adjustment, but after a few months' operation practically all the stretch will have been taken out of them, after which slight trouble will be experienced.

As the standing-rope and running-rope systems comprise considerable mechanism, it has been thought that some sort of magnetically operated device could be used that would be far more simple and compact, and certainly easier to handle, as a small electric switch could be substituted for the operating lever. This would be a decided gain in buildings where many passengers are carried, since considerable room is required for the proper manipulation of the operating lever, while for the movement of a small electric switch very little space is needed. Another advantage would be that the ropes and sheaves, which take up quite a lot of space, would be replaced by small wires which could be located in any convenient position, the connection with the car being made through a flexible cable similar to that used for conveying current for incandescent lamps.

Many inventors have endeavored to produce an electric controller, but up to the present no one has brought out anything that is of sufficient merit to enable it to replace the mechanical devices generally applied to first-class elevators in large office buildings. The principal objection to magnetic controllers is that they do not enable the operator to control the movement of the car as closely as can be done with either the standing-rope or the running-rope method. Another objection is that they are more liable to get out of order. For these reasons, then, these controllers are not generally used for passenger elevators in large buildings. Nevertheless, they have a field of their own in which they are decidedly superior to mechanical devices, and in which they have practically superseded all other methods of control.

In many buildings owned by insurance companies and other large corporations, elevators and dumb-waiters have been installed for the exclusive use of officers or employees. Such elevators are not in constant use, and therefore are not provided with operators, but each passenger operates the elevator himself. As everyone who desires to use the car cannot be expected to be an experienced elevator operator, it is necessary to provide a controller so simple that anyone can use it, and this simplicty can be easily obtained in the magnetic controllers, which require only a switch or push-button at each floor, and similar devices in the car.

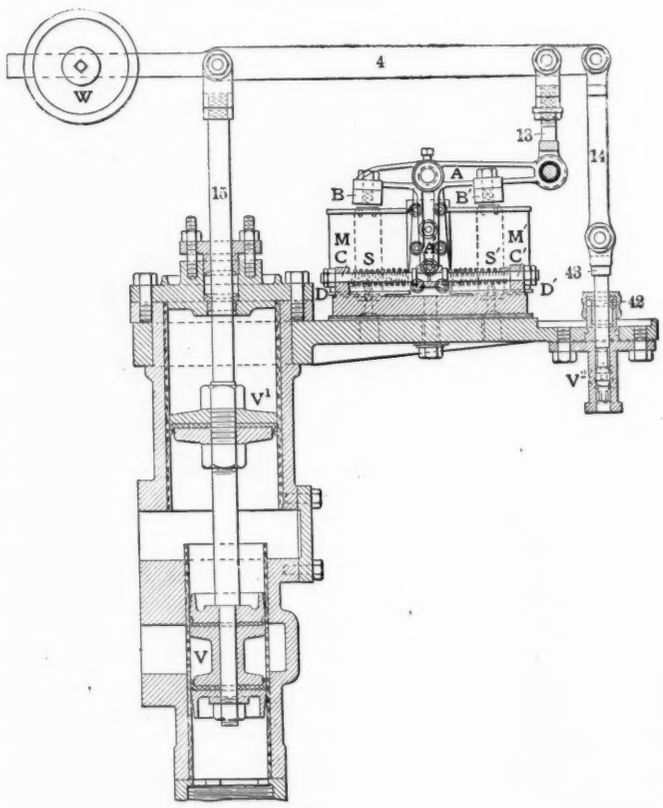


FIG. 53

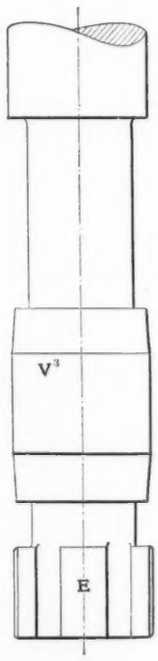


FIG. 54

Magnetically Controlled Pilot Valve

A magnetically controlled pilot-valve gear, such as is used under such circumstances, is shown in Fig. 53. The operation of this device is as follows: The

lever *A* is actuated by the pull of the magnets *M* and *M'*, which attract the armature *B* and *B'* when an electric current is passed through the magnet coils. The arm *A'*, extending downward from *A*, is held between the two horizontal rods *C* and *C'*, which are pressed against it by the springs *S* and *S'*. The nuts *D* and *D'*, together with the threaded joint in rod *13*, comprise means for adjusting the position of the pilot valve, the rod being set when the mechanism is put together, and the nuts being used for the final adjustment.

The magnets are of the horse-shoe type, there being two coils each in *M* and *M'*. The armatures *B* and *B'* are long enough to cover both poles in the magnets. By comparing Fig. 53 with the mechanically operated pilot valve which was described in April POWER, it will be seen that the magnets *M* and *M'* displace the rope sheave, and the lever *A* displaces the crank on the end of the sheave shaft. In all other respects the two arrangements are the same; the pilot valve, however, is made solid in the magnetically operated system, instead of being provided with cup packings. This difference is made in order that the valve may move more easily, and therefore not require the use of excessively large magnets. This solid pilot valve is hardened and ground to insure a perfect fit. Its actual construction is not clearly shown in Fig. 53, but will be fully understood from the enlarged illustration, Fig. 54. The lower end *E* is of the same diameter as the body of the valve V^3, in order to prevent springing and the consequent wearing of the edges of the port recessed in the valve chamber. To provide an opening for the water to escape through the lower end of the chamber into the discharge tank, the enlarged end *E* is cut away at four places, as clearly shown.

As the pull of the magnets is limited, it is necessary to guard against the entrance of chips or other solids into the valve, because these might be caught between the latter and the edges of the ports in the valve chamber and lock the valve. For this purpose strainers are provided through which all the water passes to the pilot valve. The way in which these strainers are connected, and the general arrangement of the entire valve-actuating piping, are fully shown in Figs. 55 and 56. Two strainers are provided, so that it may not be necessary to stop the elevator to clean them. Only one of the strainers is in use at a time, the two three-way cocks being provided for the purpose of connecting either strainer with the piping. The one out of service can be opened and cleaned at any time. The lower end of the bottom three-way cock is connected with the pressure tank through the pipe *K*, Fig. 55, and the lower end of the pilot-valve chamber is connected with the discharge tank through the pipe *J*, Fig. 56. The rest of the piping is so clearly shown as to require no explanation.

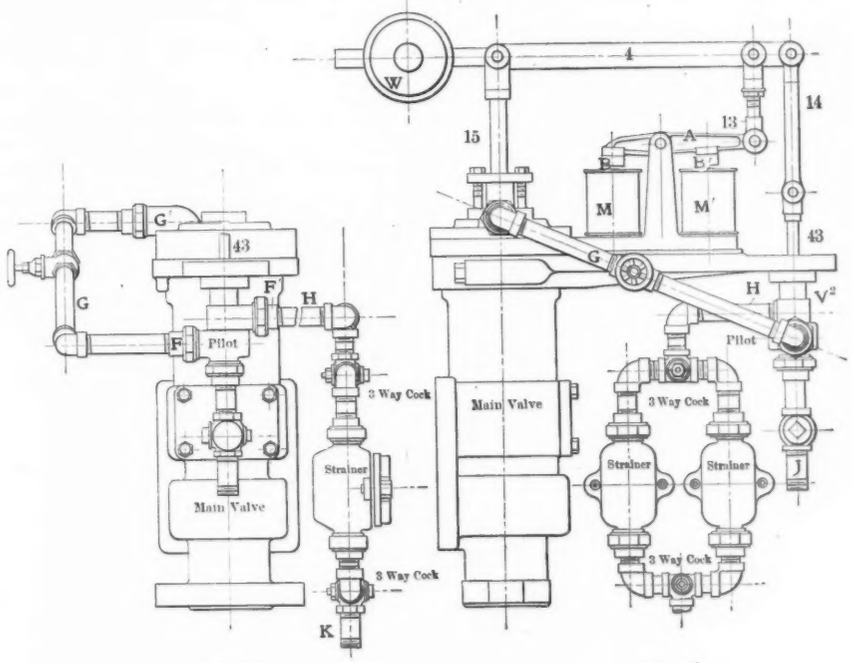


FIG. 55.

FIG. 56

MAGNETIC CONTROLLER FOR BATTERY CURRENT

The controller shown in Fig. 53 is intended for use in buildings where there is a direct-current incandescent-light circuit; but in cases where such a circuit is not available the magnets must be operated by current derived from primary or storage batteries, and as such currents are expensive it is necessary to modify the construction of the controller so that it may be operated with less current. The way in which this is accomplished is shown in Fig. 57, which shows the Otis magnet controller for battery current. This controller is also used for alternating currents, the magnets being of the solenoid type. The principal difference between Figs. 53 and 57 is that in the latter are two secondary pilot valves *D'* which are much smaller than the regular pilot valve and therefore can be moved with much less effort. These valves control the flow of water into and out of the cylinder *Z*, so as to push the piston up or down as may be required. The movement of the piston, through the rod *13* and the lever *4*, actuates the main pilot valve. The weight *W*, in both Figs. 53 and 57, is for the purpose of balancing the weight of the lever *4* to the right of rod *15*, and also the pilot valve and connection rods. The weight *W'* in Fig. 57 is for the purpose of counterbalancing the weight of *W*, the lever *4* and the pilot valve and connections.

The construction of the secondary pilot valves *D'* is not clearly shown in Fig. 57, but can be understood from Fig. 59, which is a section through the valve chambers taken at right angles to the view in Fig. 57, and also from Fig. 58, which is a plan view of Fig. 57 and shows the position of the two independent pairs of magnet spools *M M* and *M' M'*. In Fig. 58 it will be seen that the two pairs of solenoid magnet coils act independently of each other on the two valves *D'* and D_2, and when they are not energized the valves rest in the position in which they are drawn in Fig. 57, so as to connect the ports *a* and *b*. The ports *a a'*, as shown in Figs. 59 and 60, are connected with the opposite ends of the auxiliary cylinder *Z*, while the ports *b*, Fig. 59, are connected through the central passage in the valve casting with *a''*, and thus with the discharge pipe. If the magnets *M M*, Fig. 60, are energized, the valve *D'* will be drawn down and then port *a* will be connected with *c*, which, as shown in Fig. 60, is connected with the supply pipe; therefore, pressure water will pass to *a* and thus to the upper end of *Z*, and force the piston down against the tension of the spring *S'*. Thus the pilot valve *43* will be depressed and the pressure water will pass through the pipe *E*, Fig. 60, to the upper end of the main valve chamber.

If the magnets *M' M'*, Fig. 58, are energized, the valve D_2 will be pushed down, and then the pressure-water port *c* will be connected with *a'*, and thus with the lower end of *Z*, thereby forcing the rod *13* upward against the tension of the spring *S'*, Fig. 57. This spring is made stiff enough to force the compression rings *Q Q'* against the stops and thus bring the pilot valve to the central position whenever the current is cut off from either pair of magnets. The sleeves *P P'* limit the movement of the rod *13* and thus the stroke of the pilot valve *43*. The pilot valve and valves *D' D₂* are made to fit tightly by grinding, but not too tightly, so they may move as

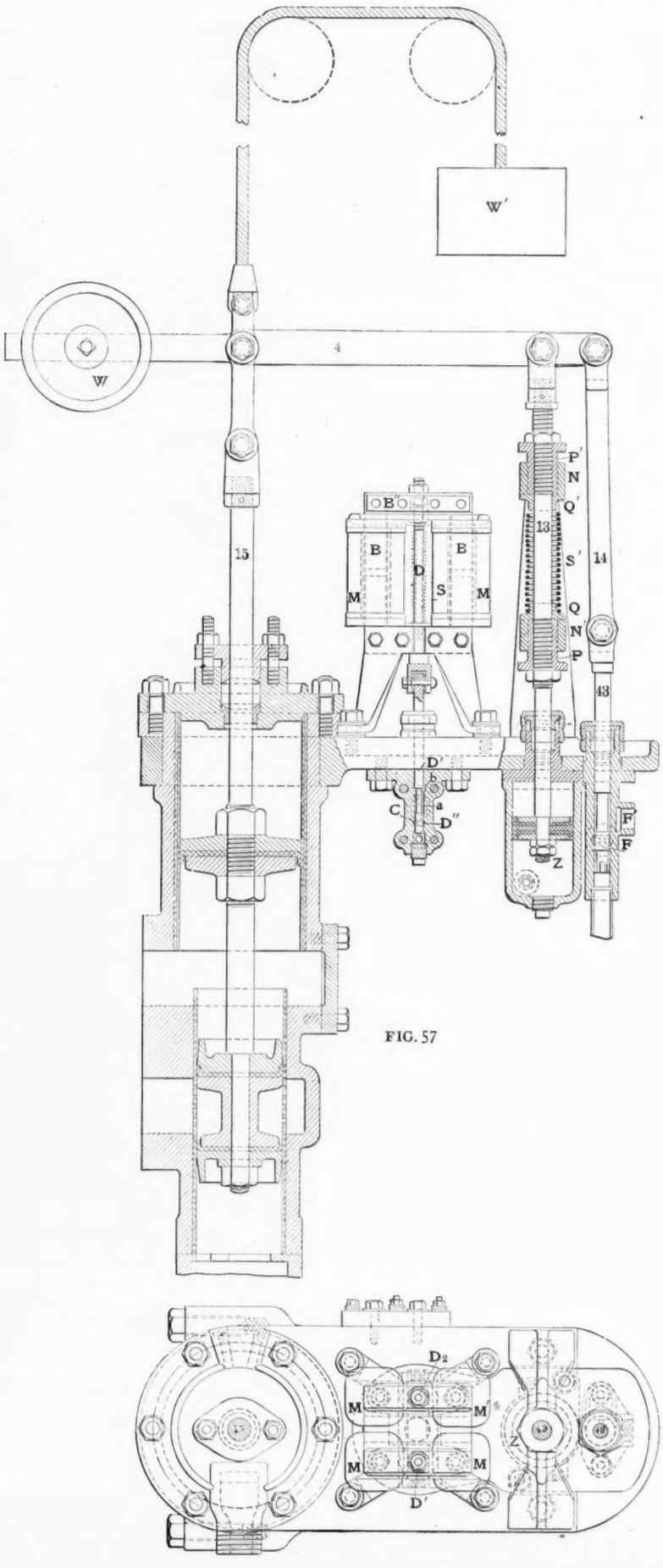


FIG. 57

FIG. 58

easily as is consistent with a good fit. The D' D_2 do not have to fit as closely as the pilot valve; for even if they leak considerably the only effect is to waste a small amount of water, because when in the stop position they both connect with the discharge pipe. Hence, after filling both ends of the cylinder Z, the additional leakage would pass to the discharge. Owing to this fact these valves can be made so as to move very easily, the principal resistance being in the stuffing-boxes; these

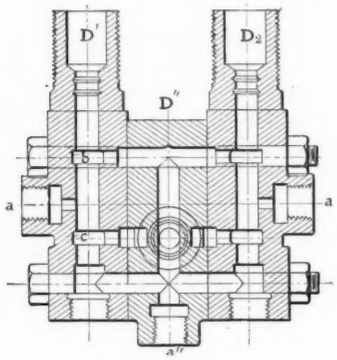


FIG. 59

should not be screwed down any more than is necessary to make them tight, and a soft packing should be used in them.

Figs. 61 and 62 are external views of the parts shown diagrammatically in Figs. 57 and 53, respectively, and will serve to show more clearly the arrangement, particularly of the pipe connections and the strainers. In Fig. 62 the two magnet spools in front are removed so as to show clearly the construction of the lever A with the extending arm A'; also the rods C C' and the tension springs S S'.

The English anti-Graft Law

The *Electrical Review* (English), commenting upon the new Prevention of Corruption act, says:

The last Christmas under the old law has passed and gone, with all its bribes and tips and presents from seller to buyer, cash, turkeys, cigars, cases of wine and miscellaneous articles, only to be enumerated by reference to a "stores" catalog. Will next Christmas be the same as its predecessors, or will the new act really see the end of these degrading customs?

And if these outward and open manifestations of the brotherly love of the man who has something to sell for the agent of the man who has something to buy really cease, will the evil simply assume new and more sinister shapes? Is it actionable to play a game of chance with the agent of a buyer and lose the game? Is it actionable to make a loan to the friend of a friend, who happens to be the agent of a buyer? Is it actionable to buy a patent from an ingenious engineer who has orders to give out?

What is corrupt consideration? Business men may well give a little thought to the point, because a mistake as to the precise meaning of those two words may decide whether some of them are inside or outside the walls of a jail next Christmas. We hope they will not feel insulted by this plain statement of fact. What we fear is that the "old hands" at the game will continue to play it in the old way, in perfect secrecy, unscathed; and that the unwary, careless, easy-going and much less criminal man may be found out in a technical offense and severely punished.

We have seen several hysterical letters, stating that it will be dangerous to tip a porter or to give "cabby" an extra twopence. Anyone who raises such a question has, of course, no notion of the meaning of the word "corruptly."

Unfortunately the new law offers no new facility for finding out; but it makes it an offense to offer or solicit a bribe. The buyer (for another party) who asks for a consideration will place himself in a very dangerous position. The buyer who even interviews a doubtful seller alone will place himself in a risky position; and we strongly advise all engineers who value their reputation to take care how they treat with the representatives of firms not known to be honest.

The fact is that, when all is said and done, the law will only be effective in so far as it is supported by public opinion. We are glad to see signs that public opinion is waking up. Chambers of commerce and many public bodies have passed resolutions favorable to the act. The local government board has issued a circular. The trades have held meetings. It is said that Lancashire coal merchants have met to consider how they are to manage without giving the customary bonus to the engineman of the mill on every ton of coal delivered.

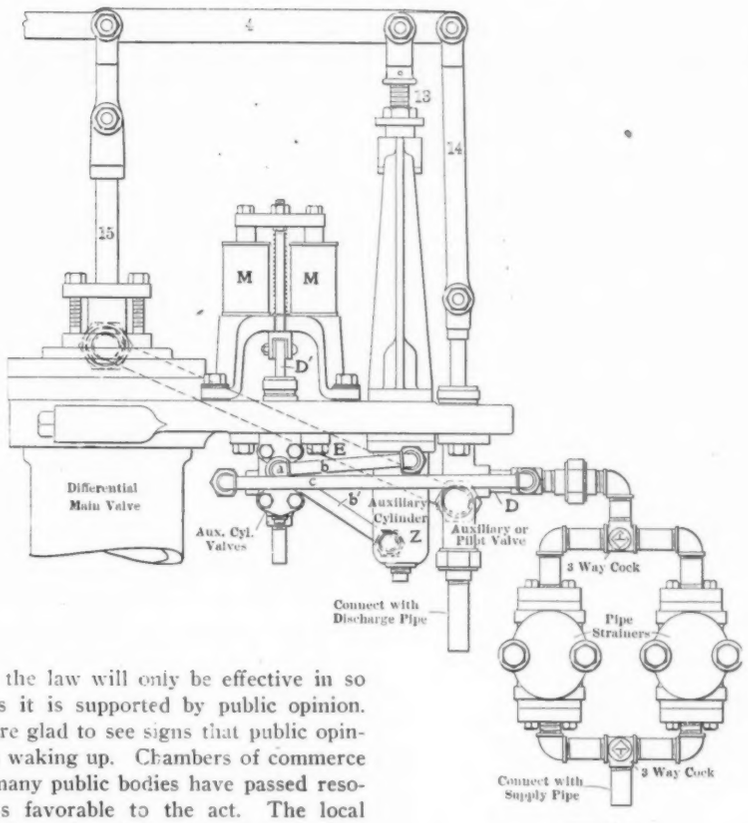


FIG. 60

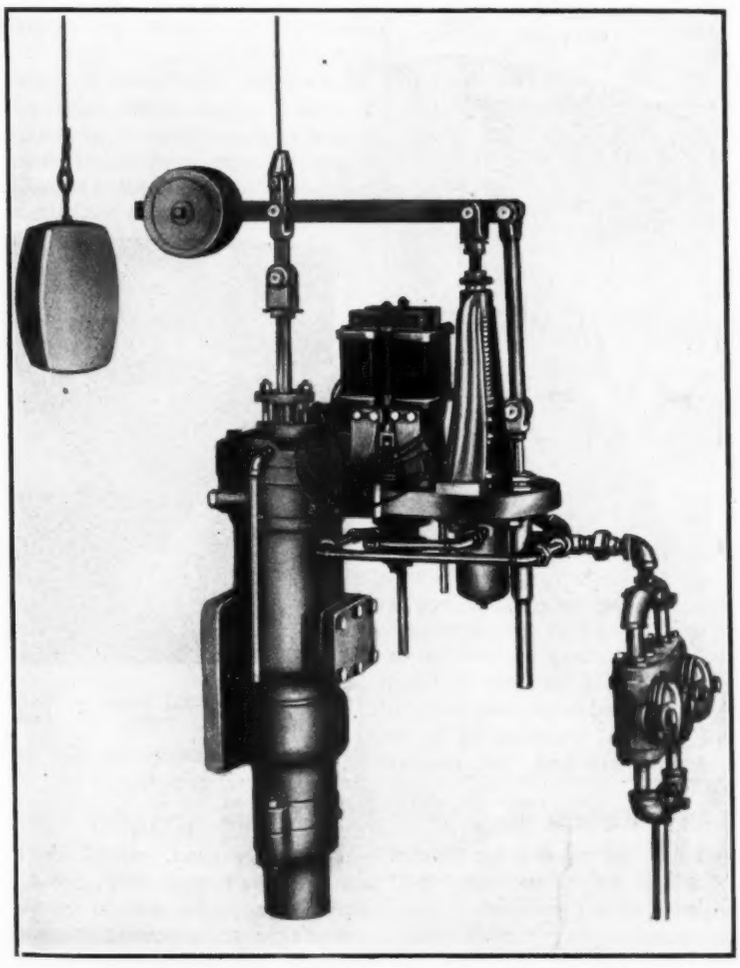


FIG. 61. EXTERNAL VIEW OF PARTS SHOWN IN FIG. 57

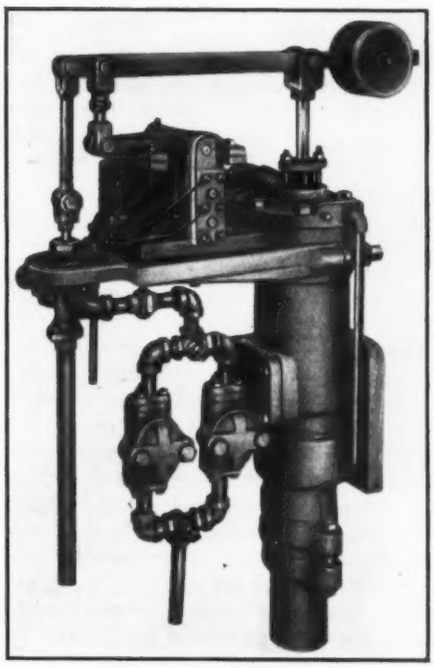


FIG. 62. EXTERNAL VIEW OF PARTS SHOWN IN FIG. 53

Perhaps the most encouraging sign of all is the formation of a league, called the Secret Commissions and Bribery Prevention League, which will institute proceedings against grafters.

Elementary Electrical Engineering—XVII

Principles of the Apparatus Employed for Starting Direct-current Motors on Constant Potential Circuits

BY CECIL P. POOLE

Motor Speed Control.

Direct-current motors, like the corresponding dynamos, are provided with field-magnet windings differing according to the class of work to be done. For most ordinary kinds of service, shunt-wound motors are used, supplied with current at

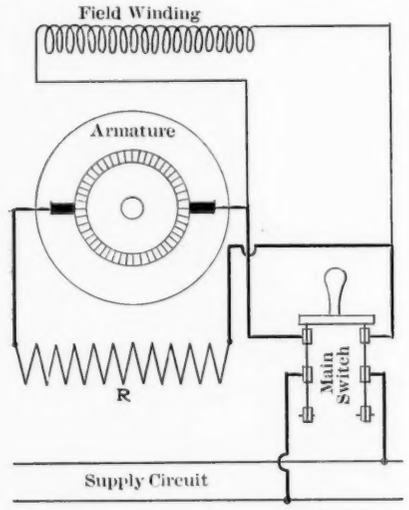


FIG. 44

constant potential. Where the motor has to start up under a heavy load, and does not need to run at constant speed after it gets under way, it is provided with a simple series field winding. When a motor must start a heavy load, and must also run at a practically constant speed after it gets started, it is equipped with a compound field winding; the series winding to give the powerful starting torque and the shunt winding to maintain a steady field after the armature has attained normal speed. The series winding is cut out of circuit when the armature has nearly reached full speed, leaving the motor running as a simple shunt-wound machine. It is the general practice now to operate motors on constant-potential circuits, partly because there are no constant-current circuits except those of very high total voltage for supplying series arc lamps and partly because the speed characteristics of a motor are much better on a constant-potential than on a constant circuit. This will be more readily understood after reading the explanation of speed characteristics in a future article.

Starting a Shunt-wound Motor

The armature circuit of a shunt-wound motor is of very low resistance, and if the machine, while at standstill, were connected directly to a constant-potential circuit of the proper voltage to operate it at normal speed, the current that would pass through would be so great as to endanger the armature winding. For example, a certain 110-volt motor had an armature resistance of 0.17 ohm, and the maximum allowable armature current was 38 amperes. Now, if that armature should be connected to a 110-volt circuit, the rush of current through it would be:

$$\frac{110}{0.17} = 647 \text{ amperes.}$$

This would obviously be dangerous to the armature windings and connections. In order to permit connection to the sup-

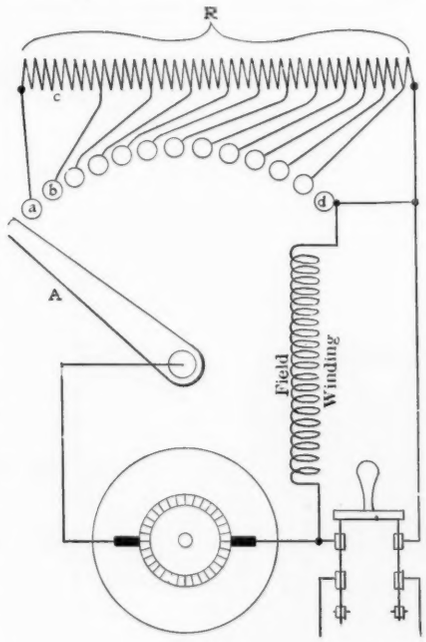


FIG. 45

ply current while the armature is at rest, an external resistance conductor is connected in series with the armature, but not the field winding. This arrangement is shown in Fig. 44, where the external resistance conductor is indicated at *R*. If this conductor be made of a resistance of 2.73 ohms, the total resistance of the armature circuit, including the external wire, will be:

$$2.73 + 0.17 = 2.9 \text{ ohms,}$$

and the rush of current through the armature, when it is first connected to the circuit, will be:

$$\frac{110}{2.9} = 37.9 \text{ amperes,}$$

which is within the maximum allowance.

When this initial rush of current occurs, it produces a strong torque in the armature, the field magnet being energized from the supply circuit, as shown in Fig. 44. If the constructional details of the machine be those specified in last month's article, the torque will be:

$$\frac{750{,}000 \times 366 \times 4 \times 37.9}{852{,}200{,}000 \times 2} = 24.4$$

pound-feet, or a torque of 0.6438 *pound-foot per ampere of current.* The armature was 7.66 inches in diameter, so that the radius was 3.83 inches or $3.83 \div 12 = 0.3192$ foot. The pull at the periphery, therefore, due to the initial current rush of 37.9 amperes would be:

$$\frac{24.4}{0.3192} = 76.46 \text{ pounds.}$$

Now suppose the load on the armature were such as to exert a backward pull equivalent to 65 pounds; this would mean a torque of $65 \times 0.3192 = 20.75$ pound-feet, which would require a current of:

$$20.75 \times 0.6438 = 13.36 \text{ amperes.}$$

The armature would speed up until its counter e.m.f. became high enough to reduce the current to that strength. The armature and external resistance equal

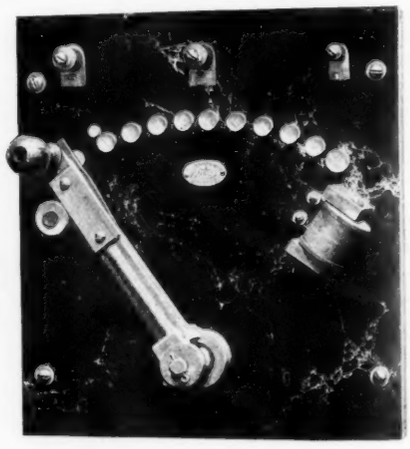


FIG. 46. COMMERCIAL FORM OF STARTING BOX

2.9 ohms; consequently the drop with 13.36 amperes passing will be:

$$2.9 \times 13.36 = 38.74 \text{ volts;}$$

the counter e.m.f. would therefore be $110 - 38.74 = 71.26$ volts, and the armature speed necessary to generate this would be 774½ revolutions per minute. The armature would accelerate until it

reached this speed, and then continue running at 774½ revolutions per minute until either the load or the external resistance were changed.

In order to get the motor up to full

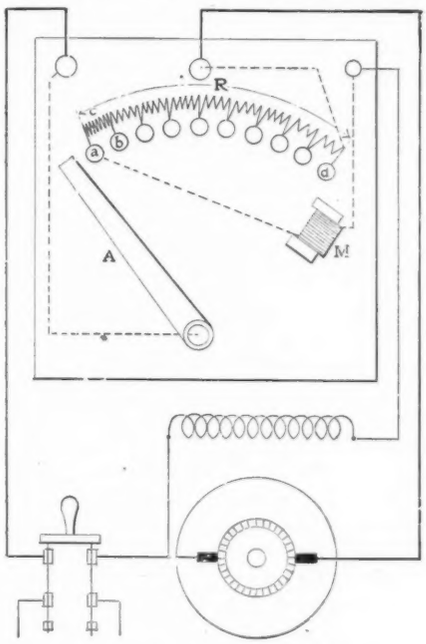


FIG. 47. DIAGRAM OF STARTING-BOX CONNECTIONS

speed, the resistance R must be cut out, but this must be done gradually. In practice the "starting resistance," as it is called, is short-circuited section by section until it is all cut out. The arrangement for doing this consists ordinarily of a pivoted lever arranged to move over a row of contact buttons or plates, as in Fig. 45, these contacts and the lever being connected to the starting resistance as indicated. When the lever A rests on the button a, all of the resistance wire R is

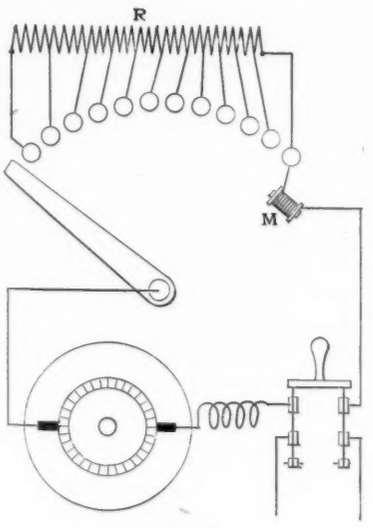


FIG. 48

in series with the armature. After the armature has started up and gained some momentum, the lever is moved to the second button b, thereby cutting out of circuit the section c of the starting resistance. This causes the armature to increase its speed still more, when the arm can be moved to the next button, and so on until it rests on the button d, which cuts out all of the starting resistance. An apparatus of this kind is a "motor-starting rheostat," but it is commonly referred to as a "starting-box," the resistance conductor usually being contained in a metal box or box-shaped frame. On one side of this structure is bolted a slate slab on which are mounted the lever and contact buttons. Fig. 46 illustrates a commercial form of starting-box for shunt-wound motors, and Fig. 47 is a diagram of its connections.

It will be noticed that a small electromagnet M is mounted on the slate front and connected in series with the field winding of the motor. The function of this is to hold the lever A in the running position as long as there is current in the field winding and to release it if that current should cease. A spring coiled around the post on which the lever is pivoted throws the lever back to the starting position when it is released by the magnet M.

Starting Series-wound Motors

A series-wound motor operated on a constant potential circuit requires a starting-box almost exactly like that used for the shunt-wound motor. The only fundamental difference is that the retaining magnet is connected in series with the motor, as shown in Fig. 48, or else across the supply circuit, as in Fig. 49. As a rule, series-wound motors are not operated constantly at maximum speed, but are regulated according to the demands of the load. In such case some special type of speed-controller is used; the commercial forms of this class of apparatus are too numerous for description here. The principal types will be described in a future article.

Compound-wound Motors

There are two general classes of com-

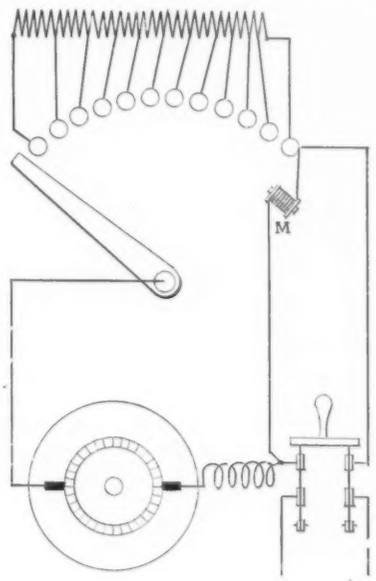


FIG. 49

pound-wound motors; one has a series field winding of very low resistance and having a much smaller magnetizing power than that of the shunt winding, and the other has a very powerful series winding of relatively high resistance. The former is provided with a starting-box exactly like that used on a simple shunt-wound motor, and its series field winding re-

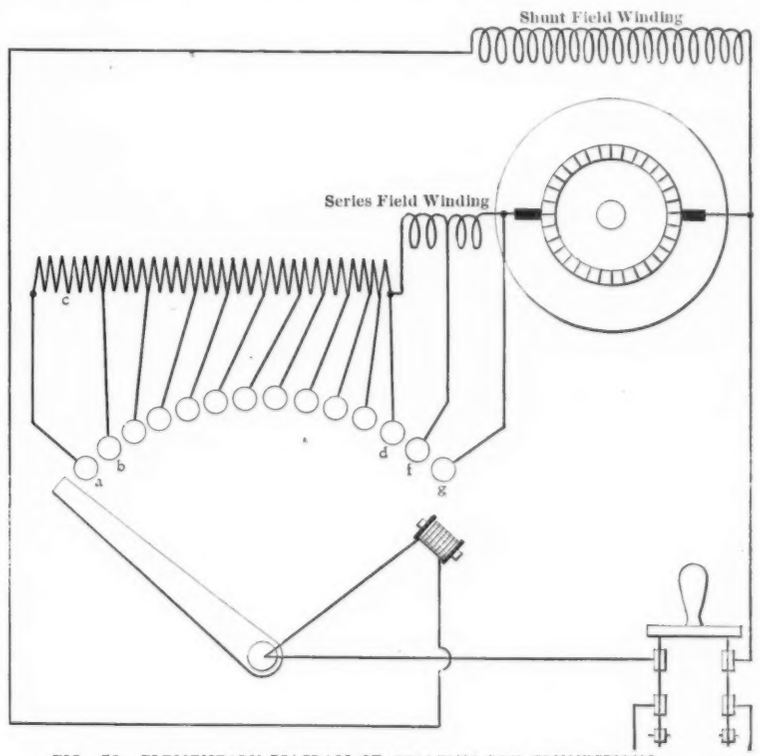


FIG. 50. ELEMENTARY DIAGRAM OF STARTING-BOX CONNECTIONS

mains constantly in circuit. The starting-box or mechanism of the second class, however, is arranged to cut out the series field windings after the external starting resistance has been cut out. The elementary diagram of connections for this class is shown by Fig. 50. As the lever or arm is moved across the row of buttons it cuts out the resistance wire one section at a time until, at *d*, all of that wire is out of circuit. Moving it to the next button *f* cuts out half of the series winding, and when it touches the last button *g*, the remainder of the series field winding is cut out and the motor runs as a shunt-wound machine. This class of motor is used largely for elevators, and the starting resistance and series field winding are cut out by special switches instead of by an arm sliding across the contacts. The switches are operated either by magnets or by a special mechanism which closes them successively and opens them in reverse succession.

Experience with a Putnam Engine

By W. H. Wakeman

The following experience with an old-fashioned Putnam engine should prove valuable to engineers in charge of any kind of a slow-speed governor that responds slowly and imperfectly to changes in the load carried, especially where electric machinery forms part of the equipment.

The conditions in this plant are as follows: A 14x36-inch engine is fitted with the governor illustrated in Fig. 1, the central spindle of which revolves 55 times per minute without tension on the spring. The bevel gear on the lower part of it has 30 teeth and meshes into one that contains 44 teeth; therefore the side shaft that is located parallel to the frame (see Fig. 2) revolves $55 \times 30 \div 44 = 37.5$ times per minute. As each of the bevel gears at *A* has 28 teeth, the large gear *B* also revolves 37.5 times per minute. It is fitted with 108 teeth, while the crank-shaft gear *C* has 54; therefore the fly-wheel revolves $37.5 \times 108 \div 54 = 75$ times per minute. This wheel is 10 feet in diameter, while the main pulley is 3 feet, and on the same shaft there is a 48-inch pulley that drives a belted dynamo, on which is an 11-inch pulley; therefore the latter revolves 1090 times per minute, provided the belts do not slip.

As this dynamo was built to run 1050 revolutions per minute, it works well when the engine is running at standard speed; but unfortunately this desirable condition can only be secured for a part of the time. The load on this dynamo

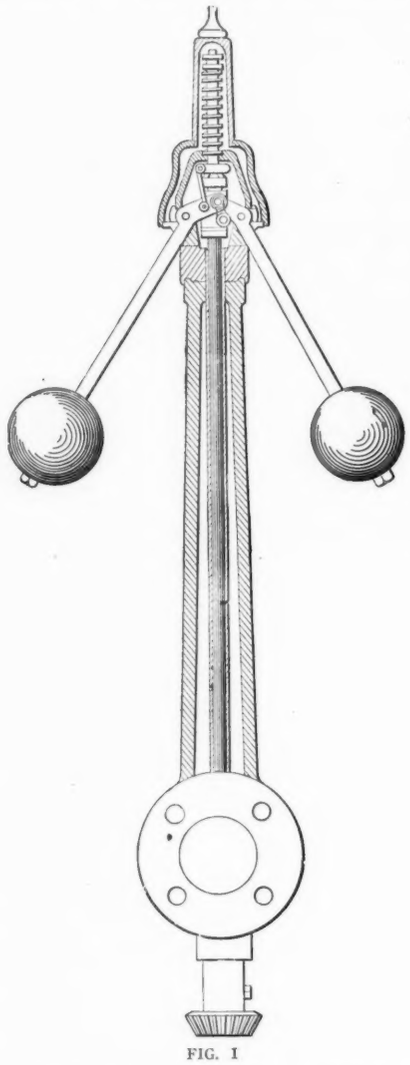


FIG. 1

varies from 5 to 150 amperes, which is its full capacity. The various machines in the shop are started and stopped frequently, and the back pressure varies from nothing to 6 pounds by the gage; therefore the load varies from about 10 to 60 horse-power. With about one-half load the standard speed was secured approximately, but when the full load was on the speed fell to 70 per minute; and as the dynamo revolves 14.5 times for each revolution of the engine, this reduced the speed of the former 72.5 turns per minute. When the boiler pressure was lowered for any cause, it was even less than this indicates; consequently, with the rheostat wide open it was not possible to secure full potential on the lamps, and under these conditions the armature and brushes heated badly, so that there was danger of damaging the dynamo. Brushes were very nicely fitted to the commutator, and other remedies for heating were applied, but the temperature remained too high.

It is necessary in such a case to keep within two extremes of operation, for with the highest boiler pressure carried and the lightest load on the engine (including no back pressure), the potential must not exceed 115 volts with the rheostat open two or three points. This is necessary in order to provide a margin for emergencies, for when the other load is thrown off suddenly, and the engineer is not at hand to close the rheostat quickly, the engine speed is increased rapidly for a short time above the normal or standard, causing the voltage to go high enough to melt the connections to the voltmeter. The other extreme consists of having speed enough while carrying the highest boiler pressure, with a full load on the engine, to give 115 volts potential at the switchboard before the rheostat is wide open, thus making an allowance for a few pounds less boiler pressure.

With a belt-driven governor it is an easy matter to secure more speed by wrapping one or more turns of leather belting around the governor pulley, thus increasing its diameter until the required speed is found; then a new iron pulley should be made and put on; but with a gear-driven governor the case is quite different. However, it is possible to secure a few more revolutions of this governor by means of the spring shown in the upper part of it in Fig. 1. The nut which holds this in place was screwed down until the speed of the engine was increased 5 revolutions per minute, and fastened in place by a check-nut. Of course, this caused all of the machinery in the shop to run faster, but it did not prove objectionable, and it has made a great improvement in some respects. With a

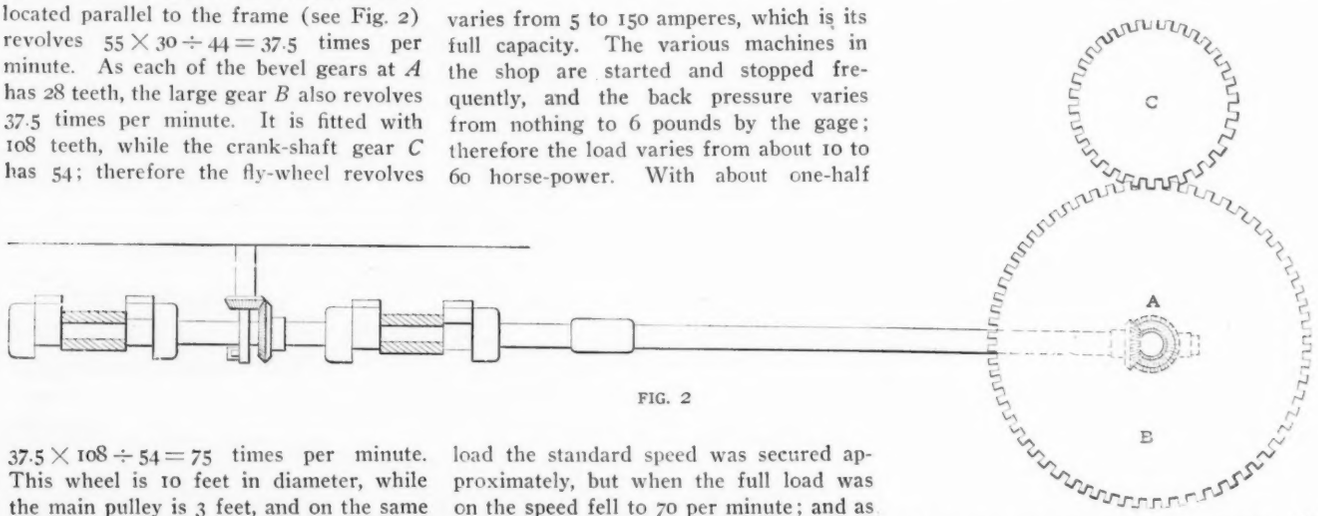


FIG. 2

light load and a high boiler pressure the rheostat can be used in the position shown in Fig. 3 without exceeding the required potential of 115 volts, and with a heavy load it is frequently found as illustrated in Fig. 4, even with 5 pounds less than full boiler pressure. Furthermore, every movement of the indicator has more effect on the voltage than before; or, in other words, it is "smarter" and "answers

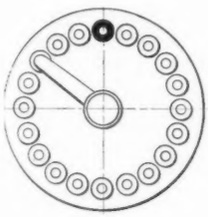


FIG. 3

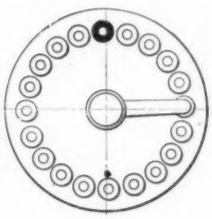


FIG. 4

to the helm" better. The commutator and brushes run much cooler than formerly, although the speed is 72.5 revolutions more per minute, and 112.5 higher than it was designed for. The latter improvement is highly appreciated, and the same remedy may apply to other cases where excessive heating causes trouble and annoyance.

The double-beat poppet valves of this engine are illustrated in Fig. 5. After these valves had been used 12 years, there were slight leaks around the valve-stems

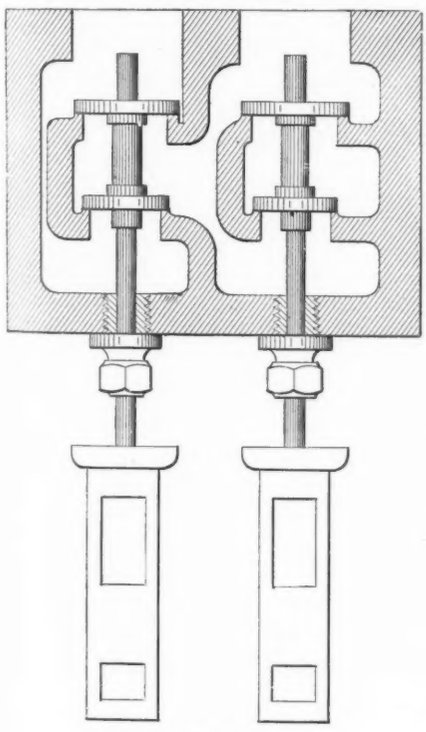


FIG. 5

as they were worn where they work through the packing. They were put in a lathe and a chip 1/64 inch deep turned off from them, which proved sufficient to make them perfect. This is mentioned because it represents very slight wear for such a long time. They have never been packed with anything else but asbestos wicking coated with graphite mixed with a nearly pure mineral cylinder-oil, which has always been renewed semi-annually without regard to its condition, when it was time to remove it. These valves will work quietly as long as the stems are not worn enough to cause them to bind in the stuffing-boxes, when an effort is made to prevent leakage at these points by tightening the nuts; therefore, when they begin to be noisy after being packed as above described, it is time to have a chip taken off from the stems; but care should be taken to remove no more stock than is actually necessary.

Spring Meeting of the American Society of Mechanical Engineers

The spring meeting of the Mechanical Engineers will be held this year at Indianapolis, commencing May 28. The Claypool hotel will be the headquarters and the professional sessions will be held in the auditorium of that house.

Tuesday evening will be devoted to the hearing of the president's address and the informal reception which, by custom, follows.

Wednesday forenoon will be devoted to a business session, when in addition to the routine work of the society reports will be received and considered from the committees on Standard Proportions for Machine Screws, and Testing Refrigerating Machines. Time permitting, the following papers will be presented at this session:

"Collapsing Pressures of Lap-welded Steel Tubes." By Professor Reid T. Stewart.

"The Balancing of Pumping Engines." By A. F. Nagle.

"A Comparison of Long and Short Kilns." By E. C. Soper.

Wednesday afternoon will be given over to visits to various plants.

The Wednesday evening session will consist of an automobile symposium, for which the following papers are already in hand:

"Bearings and Moving Mechanism." By Henry Hess.

"Air Cooling of Automobile Engines." By John Wilkinson.

"Materials for Automobiles." By Elwood Haynes.

"Special Auto Steel." By T. J. Fay.

"Railway Motor car." By D. B. Gray.

It is hoped that this may lead to the formation of an automobile section, it being the belief of the leaders that if there is a call for an organization devoted to the mechanical side of the automobile industry, it will be better filled by an organization within the society than by a new and separate association.

Thursday and Friday forenoons will comprise a symposium on "Superheated Steam," with the following papers:

"The Specific Heat of Superheated Steam." By A. R. Dodge.

"Determination of Entropy Lines for Superheated Steam." By A. M. Greene.

"The Flow of Superheated Steam in Pipes." By E. H. Foster.

"Co-relation of Furnace and Superheated Conditions." By R. P. Bolton.

"Performance of Cole Superheaters." By W. F. M. Goss.

"Experiences with Superheated Steam." By G. H. Barrus.

"Use of Superheated Steam on Locomotives in America." By H. H. Vaughan.

"Superheated Steam in an Injector." By S. L. Kneass.

Professor S. A. Reeve has a paper entitled "A Hirn's Analysis of Locomotive Test," and H. O. Lacount one on "The Heating of Storehouses," and other papers have been promised but are not in hand at this writing.

Thursday afternoon will be devoted to an excursion and Thursday evening to a reception. Friday's session will be held at Purdue University at Lafayette. An invitation is extended to visit the University of Illinois at Urbana, where Dr. L. P. Breckenridge will receive and entertain those who avail themselves of the invitation.

As this issue goes to press May 15, we are obliged to defer any attention to the papers until our July issue.

Reading Technical Papers

Where a number of technical papers are to be read each week or month, it becomes essential to learn how to get the cream without drinking all the pans of milk. It is out of the question to read them all thoroughly. Before attempting any reading an engineer should look them all over and note important articles to be read later if possible. In this way there is obtained a general idea of the contents of the papers and also of the relative importance to him of the articles noted. These are then carefully read in the order of merit, so far as time will permit. In this way one will see and read many things of the first importance, whereas without such a preliminary glance through the papers they might have been missed. Without going through papers it cannot be known what they contain; while to begin and read as you go, results in using up all the time over matter which may be of little relative importance. One must, however, avoid the danger of being satisfied with the preliminary skimming; the thorough reading must follow.—H. Wade Hubbard.

First Rateau Regenerator Installed in America

Description and Test of Rateau Low-pressure Turbine System at the Works of the International Harvester Company

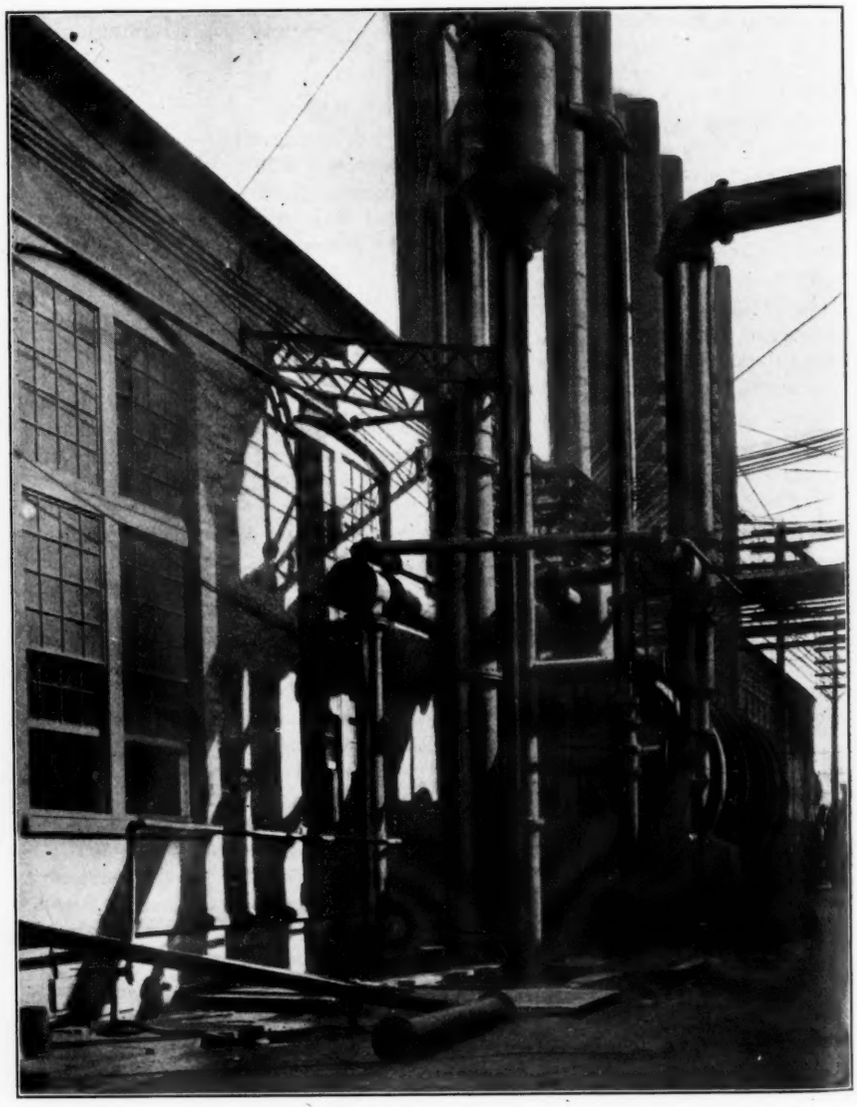
BY F. G. GASCHE

The International Harvester Company, of Chicago, Ill., has recently completed the installation of the first complete equipment of low-pressure steam turbine and regenerator on the system devised by Professor Rateau, of Paris. Notwithstanding the unquestioned success of the many European installations, American iron and steel manufacturers have been inexcusably tardy in appropriating this efficient means of reducing the cost of operation of rolling mills, particularly of the reversing type.

The writer was retained by Samuel Hale, general superintendent of the International Harvester Company's plant at South Chicago, to investigate the Rateau system, and is indebted to A. U. Leonhauser, the chief engineer of the plant, and his organization, for the exceptional facilities provided for the testing operations. Much valuable assistance was also rendered by C. J. Bacon, of the Illinois Steel Company, in the preparation of diagrams and the derivation of results, particularly with reference to the mill tests.

The excellent adaptation of the means to the end is due to the efforts of Mr. Leonhauser, in collaboration with C. H. Smoot, of the Rateau company. The contract for the leading elements of the plant was assumed by the American Rateau Steam Regenerator Company, 138 Jackson boulevard, Chicago, which controls the exclusive American rights of Professor Rateau's regenerator patents. The Rateau company supplied the following parts of the plant, the remainder being provided by the International Harvester Company, viz.:

The steam regenerator, or accumulator, consisting of a cylindrical wrought-steel shell ⅜ of an inch in thickness, 11 feet 6 inches in diameter, and 30 feet long, having a central horizontal diaphragm which divides the regenerator into two similar compartments. In each compartment there are six elliptical tubes or steam-distributing conduits, *A*, Fig. 2, which extend from end to end in pairs, and are so placed as to leave spaces, *B*, between them. (The sectional view is from another installation and only shows four tubes.) Baffle plates, *C*, are arranged above the space between each pair of tubes. The spaces surrounding the conduits, and, under certain conditions, even the conduits themselves, are filled with water to the extent that the top of the latter is usually submerged three or four inches. The sides of the conduits are perforated with a great many ¾-inch holes to allow of the lateral escape of steam through the water with, occasionally, a further escape from the bottom openings. A large baffle plate in the upper steam space serves for a perfect separation of entrained moisture from the steam. The steam enters by the pipe shown at the left hand of the side elevation, passes to the interior of the elliptical tubes, and escapes into the spaces through the perforations. The circulation of the water takes place in the direction of the arrows; the baffle plates placed above each pair of tubes prevent the water from being thrown into the steam space. This flow of steam gives an extreme degree of steam saturation to the water; and the slight back pressure which at first sight might be expected, owing to the head of water above the rows of perforations, is thereby reduced to insignificant proportions.



RATEAU REGENERATOR AND ATTACHED CONDENSER

When the supply of steam from main engine ceases, the water liberates part of the heat it has absorbed, and an even flow of low-pressure steam is given off, while the steady demand of the turbine reduces the pressure in the accumulator,

causing the steam still retained in the tubes to escape, maintaining the circulation of the water, and facilitating the liberation of the steam. Experience has shown that the whole of the contained water participates in the regenerative action. The steam is taken from the top of the accumulator to the turbine, and the pressure can be regulated by the relief valve shown. The water level is maintained constant by a ball-float contained in a small tank arranged at the back of the regenerator. Generally there is a slight overflow at all times, representing among other things the "makeup" from the exhaust-steam supply.

The regenerator at this plant has a capacity of 55 tons of water, sufficient by actual test to deliver all the steam for a 50 per cent. load on the turbine for a period of 430 seconds; at full load this would correspond to a period of 390 seconds. The adaptation of the capacity of the regenerator for a given case is, at least commercially, a local problem.

Accessories

The accumulators are fitted with the following accessories:

Adjustable relief valve. This valve is fitted to regulate the limits of pressure in the accumulator, and to allow the steam to escape when the turbine is stopped or working on a light load. It also insures the absence of undue back pressure in the cylinders of main engines. This valve may be connected to the condenser so that if the turbine is standing the main engines may have the benefit of the vacuum, and in cases where surface condensers are installed, pure hot water will be saved for use in boilers.

Non-return water valve. This valve is generally necessary with water accumulators, to prevent any possibility of reflux of water toward main engines during long periods of stoppage. It is placed in a by-pass connecting the steam reservoir of the regenerator and the exhaust pipe of the main engine.

Automatic level regulators and gage glasses are fitted to water accumulators, and automatic drains to mixed or scrap type accumulators.

Piping. Beginning at the inlet of the receiving drum, including the steam header and mains from the regenerator to the turbine, the exhaust piping from the turbine and condenser, and the piping between the condenser and the air pump.

A vertical receiving drum 9 feet in diameter and 22 feet long, with baffle plates and separating chambers, the function of which is to allow the ready escape of steam from the mill engine without increase of back pressure on the system. The expansions allowed in this drum conduce to a more even flow of steam in the steam regenerator.

A 500-kilowatt turbine generator set mounted on a substantial base plate, consisting of a Rateau low-pressure turbine and two similar direct-current generators each of 250 kilowatts capacity, at 250 volts, designed so that they may be operated in parallel. The bearings are of the ring-oiled reservoir type, with water jackets.

A 30-inch barometric condenser of the Alberger type complete with air cooler, exhaust entrainer, expansion joint, and an air pump 8x16x12 inches.

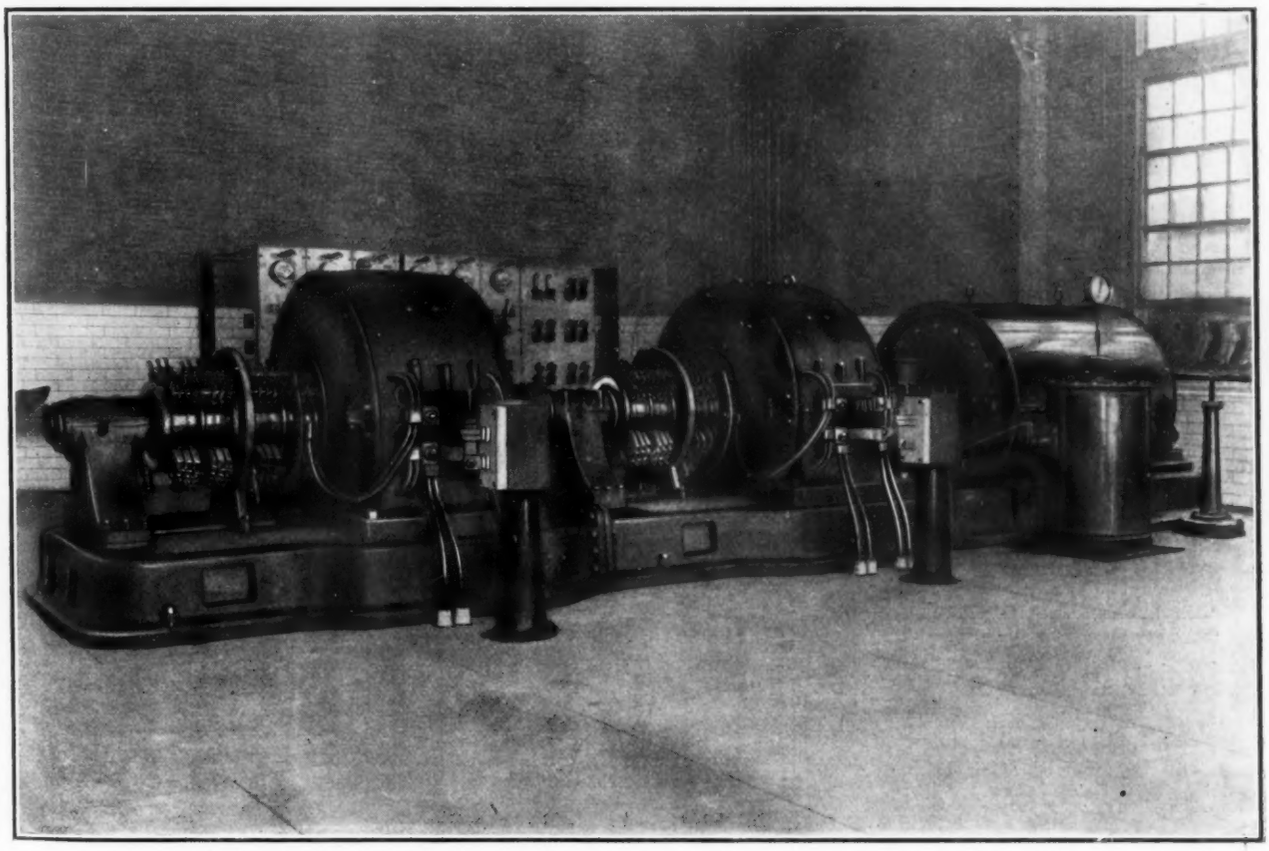
The plant is designed with the view of adding another similar unit, but the evidence of the tests shows that a 750-kilowatt unit can be operated with the steam that is available, without allowance for the steam (about 6000 pounds per hour) that is available from auxiliary machinery.

Part of these auxiliaries already deliver into an open feed-water heater, but the steam regenerator, constituting a perfect feed-heating device, can more appropriately receive all the steam from auxiliaries with the advantage of some addition to the capacity of the turbine equipment.

General Methods of Testing

The leading objects of the investigation were: first, to determine the steam consumption of the turbine per unit of power; and, second, to measure the actual amount of steam available for the use of the turbine as delivered from the main engine.

The estimation of the steam requirements of a reversing rolling-mill engine has heretofore been a matter of the greatest difficulty; always subject to grave errors. On the occasion of these tests exceptional facilities were provided for



RATEAU LOW-PRESSURE STEAM TURBINE AT INTERNATIONAL HARVESTER COMPANY'S WORKS

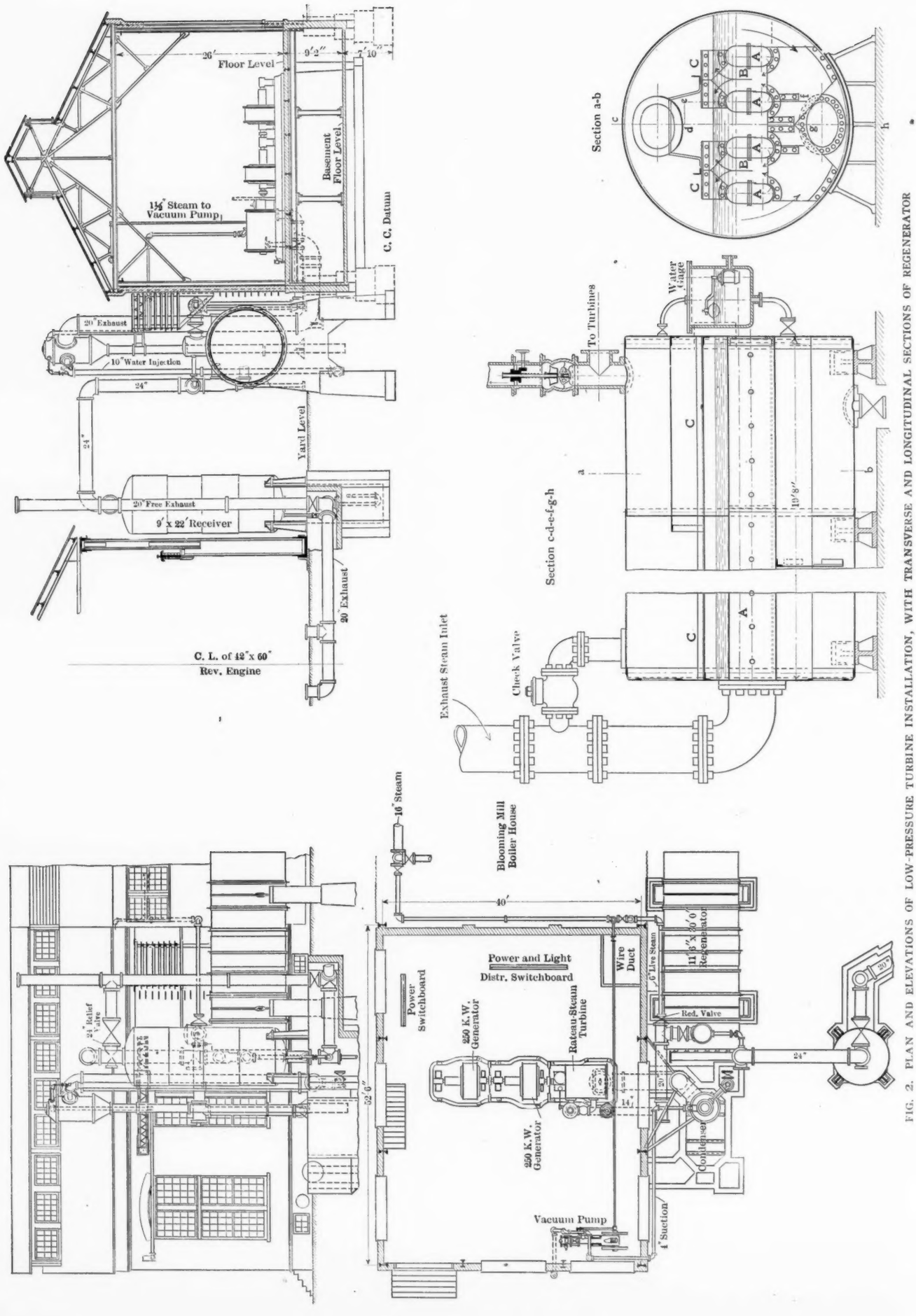


FIG. 2. PLAN AND ELEVATIONS OF LOW-PRESSURE TURBINE INSTALLATION, WITH TRANSVERSE AND LONGITUDINAL SECTIONS OF REGENERATOR

making this a direct and positive measurement. That a proper interpretation of the results might be possible, and to place the information in such form that comparisons with other rolling mills can be made, it is essential to specify the mill condition by many observations concerning the roll train and the several passes in the rolls.

In testing the turbine and generators, the steam consumption at four different load conditions was measured, by the following process:

A part of the exhaust steam from the main engine was passed into the steam regenerator, through the system of exhaust piping and the expansion chamber. The excess steam escaped through a partially opened by-pass valve and partially through a Shutte & Koerting back-pressure valve, adaptable to low pressures. When for any reason the operation of the main roll-train engine was interrupted for a period of more than three minutes, steam was automatically admitted to the regenerator from the live-steam mains through a Davis pressure-reducing valve. This insured a continuous flow of steam to the turbine during the test at pressures that deviated but a few tenths of a pound from the average regenerator pressure. As the turbine and generators were subject to an extremely uniform load, due to the use of a large water rheostat, they alone being under examination at the time, there was every excuse for the occasional admission of live steam to the regenerators.

The barometric condenser was appropriated as a large calorimeter to measure the total heat rejected from the turbine and to ultimately determine its steam consumption at the given load. The water supplied to the condenser was measured by inserting a Venturi water meter in the injection pipe. The difference of head between the "up-stream" end and the "throat" was measured by a closed mercury column. Since the throat was 5 inches in diameter, and the up-stream diameter 10 inches, the velocity through the throat of this particular meter was expressed with an error probably not exceeding 1½ per cent. by the formula:

$$v = 29.72\ h^{\frac{1}{2}},$$

where h = the mercury column in feet, and v = the velocity in feet per second.

Temperatures of injection water and of the barometric column were determined by thermometers calibrated by a Henry Green "Standard," in the possession of the South Works of the Illinois Steel Company.

The total steam delivered to the turbine is readily estimated from the formula:

$$w = \frac{W(T'' - T') + Q + Q' + H}{Lx + q(1 - x) - (T' - 32)},$$

where:

w = pounds of steam delivered to the turbine per minute,
W = pounds of condensing water per minute,
L = latent heat of steam as supplied to turbine,
q = "heat of liquid" corresponding to L,
T' = injection-water temperature,
T'' = hot-well temperature,
Q' = radiation from exhaust piping, B.t.u. per minute,
Q = radiation from turbine and steam connections, B.t.u. per minute,
H = heat equivalent of work done at turbine shaft in B.t.u. per minute.

The value of x, the quality of steam as supplied to the turbine, was taken as unity, as a throttling calorimeter attached to the steam pipe near the turbine and exhausting into the condensing system showed dry saturated steam coming from the regenerator.

Q and Q' were estimated from temperature ranges and consideration of the surfaces involved, using the value quoted by G. H. Barrus, M.E., for a bare 10-inch pipe surface (see A. S. M. E. *Proceedings*, 1902, page 832). This is approximately 3.2 B.t.u. per hour, per square foot and per degree of difference of temperature. Radiations thus found by Barrus apply to a steam pressure of 150 pounds, while in the present application the maximum temperatures seldom exceed 220 degrees Fahrenheit. In all probability the radiation loss as applied to the turbine tests is somewhat over-estimated. The correction for radiation being of small influence, scarcely 1 per cent., it is best to adhere to measured values as a guide.

Knowing the electrical energy at the switchboard and the mechanical efficiency of the turbo-generator set, it is easy to estimate the value of H, since one horsepower at the turbine shaft has a heat equivalent of 42.416 B.t.u. per minute.

The measurement of steam delivered from the 42x60-inch reversing engine became a possibility, owing to the presence of the regenerator to even the flow to the condenser. For the purpose of this test the exhaust pipe from the turbine was disconnected, and a new pipe inserted between the regenerator and the condenser. A small air relief-valve was attached to this pipe so as to prevent the formation of a vacuum in the regenerator.

The injection water was limited to the extent that the hot-well temperatures were 139.2 degrees Fahrenheit, and 132.1 degrees Fahrenheit for tests 1 and 2, respectively. With the belief that the air entering the exhaust system, though apparently small in amount, could be almost completely eliminated as a heat-absorbing agent, it was thought well to maintain the higher temperatures of 183.6 and 174.2 for tests 3 and 4, respectively. Under these circumstances there was seldom an observable lifting of the air relief-valve, when there was anything like a normal operation of the rolling mill.

Of course, the entire steam delivery from the main engine was passed into the regenerator and its perfect regulation of flow into the condenser rendered conditions so uniform that considerable accuracy in determining the rate of steam flow from a mill reversing engine was secured. Bearing in mind that we had already found a dry and saturated condition of the steam coming from the regenerator, whence $x = 1$, the formula for calculating the steam delivery per minute becomes:

$$w = \frac{W(T'' - T') + Q'}{L - (T' - 32)}.$$

For the turbine test the above described values were used except that of Q'', which is estimated at 84,000 B.t.u. per minute, in view of the large extent of surface at present unprotected by any

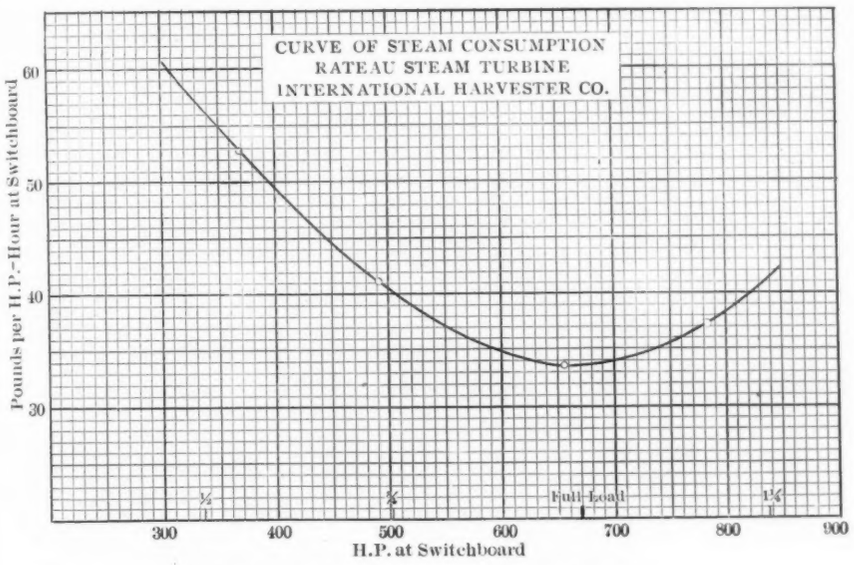


FIG. 3. CURVE OF STEAM CONSUMPTION OF TURBINE

covering. *H*, of course, is zero in this application of the condenser. This process leads to a statement of the amount of steam available for the purposes of the turbines. When estimating the total steam consumption per horse-power of the reversing engine an allowance must be made for the amount of steam appearing as "separator drip" at the expansion chamber.

With all the seeming variations of load conditions imposed by the ordinary reversing roll train, there is nevertheless, for the same class of steel product, a well-defined cycle of operations into which the mill equipment of machinery and its operatives gravitate. In the particular case under examination it became a current practice to roll a bar to a 3⅞-inch square section, approximately, from an ingot weighing 5490 pounds, the butt end of which was 17¾x20 inches.

This required twenty-one consecutive "passes" through the rolls, during a few of which the work demanded of the engine was as low as two and a half million foot-pounds. The rate of performing this work is quite variable, comparing one pass with another, but any given pass of the twenty-one operations requires practically the same time and work, as it recurs during the reduction of successive ingots.

Since the time intervals, though small, assume great importance in establishing all the characteristics of a typical "mill cycle," it was necessary to devise a tachometer with suitable electrical attachments for the automatic recording of the several important events connected with the rolling of an ingot. A description of the tachometer, together with methods followed in its use, will not be attempted, as the ultimate results are sufficiently defined in the diagram Fig. 1.

After making the indicator test of the engine, using continuous indicators, it is possible to give the full information, including indicated steam consumption for the period during which steel is passing through the rolls. The total steam consumption per hour is estimated on the basis of the number of "mill cycles" that were completed during the observations of the condenser tests.

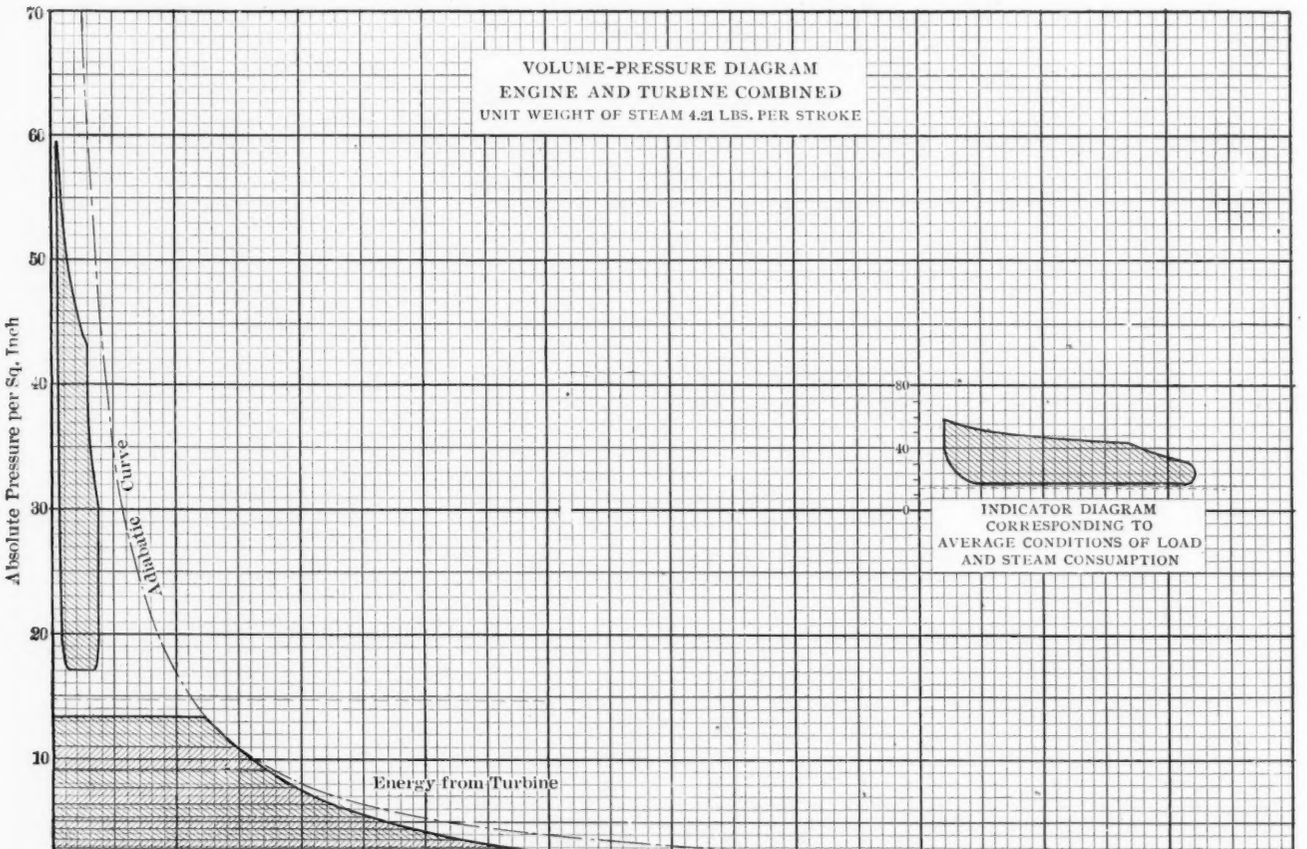


FIG. 4. VOLUME-PRESSURE DIAGRAM OF ENGINE AND TURBINE COMBINED

Test Results

Table 1 gives the average of the observed and the derived results for four different load conditions on the turbine. While a graphical exhibit of all observations of the several tests would seem to be of value, it can be asserted that the facilities for control of the water rheostat were such that all important observations would, on any legible scale, become practically straight lines. There was no occasion for adjustment of the condensing-water supply, or any other important element after a given load condition had been provided.

A shop test of the generators showed an efficiency of machine of 90.2 per cent. After a two-hour run with an 18 per cent. overload, the resistance of the water rheostat was rapidly increased with successive readings at five-minute intervals, as follows:

Time	Amp.	Volts.	R.P.M.
1.10	2750	228	1500
1.15	2750	300	1500
1.20	3000	210	1425

Immediately after this the temperatures of commutator and armature on No. 2 generator were found to be 64 and 65 degrees Centigrade, respectively, with the air temperature of 18 degrees Centigrade, making a rise of temperature of 46 and 47 degrees Centigrade for the two parts. This is well within practical limits.

Fig. 3 shows the curve of steam consumption of the turbine under actual conditions of service. The turbine was supposed to operate with 27½ inches of vacuum for all load conditions. Local difficulties with air leaks in the condensing system at the time of the tests prevented the maintenance of this vacuum, which, if existing, would have favored a

considerable reduction of steam per kilowatt-hour, particularly at the light loads. The curve should therefore not be a finality of evidence of the attainable efficiency of the machine.

Table 2 gives the results of the four condenser tests of the 42x60 McIntosh & Hemphill engine. The groups of observations applying to the several tests were selected from those periods when the mill was operating under normal conditions; that is, when rolling at the rate of from sixteen to seventeen ingots per hour. Owing to the limited steel-making capacity at the plant, it is impossible to maintain this rate of delivery of steel to the blooming mill throughout the twelve-hour turn. While this is a temporary condition at the plant, it causes mill delays, sometimes an hour at a time. The condenser combined with the steam regenerator was so extremely sensitive to these changes that injection-water temperatures arose to their normal test values within two minutes after the mill commenced a rolling period.

As a matter of great convenience the entire processes of test, both for the turbine and for the engine, consisted in a measurement of *rates* of delivery, or flow of water, steam, heat, etc. Under such circumstances, when normal conditions of tests had obtained, a few observations would theoretically be sufficient for all calculations.

That tests of the engine and mill under operative conditions were limited to 60 minutes or 90 minutes duration in no sense jeopardizes the accuracy of the ultimate results; it does, however, eliminate accidental idle periods in the mill.

During the complete mill treatment of an ingot, a period which has been designated as a "mill cycle," the engine exerts an aggregate of 11,000,000 foot-pounds, approximately, in about 200 seconds. This is a delivery of power at approximately 1000 horse-power, and is to be identified as the average horse-power, while the rolling process continues. When sixteen ingots are rolled per hour, there is approximately 20 per cent. of the total time during which the mill is supposed to be in operation when the engine is idle, or running so slowly that the work (outside of friction) is negligible.

If, as a consequence, the total work performed by the engine on sixteen ingots is assumed to require an hour for its delivery, the horse-power of the engine may be said to be "distributed over the hour." This is a particularly convenient form of statement, when the applications of the steam turbine are contemplated, as the delivery of the turbine is unquestionably independent of any ordinary mill interruptions in the presence of a steam regenerator of suitable capacity.

To illustrate its application, the 42x60-inch engine at the International Harvester Company's plant requires an average of 52,400 pounds of steam per hour, with a horse-power, distributed over the hour, of 820. This make 64 pounds of steam per hour per indicated horse-power. The same amount of steam on turbines of suitable capacity will enable the delivery of 1510 horse-power at the switchboard, at 33.6 pounds of steam per horse-power-hour. This makes the performance of the plant 2330 horse-power delivered for 52,400 pounds of steam, or a steam rate of 22.5 pounds.

If the mill is credited with the fuel value of the power delivered by the turbines, it is easy to derive the saving to the plant, resulting from the combination of simple reversing engine and turbine. Assume that a boiler horse-power costs $60 per annum. Then 52,400 pounds of steam per hour is approximately 1750 boiler horse-power, costing $105,000. With the original delivery of the engine of 820 indicated horse-power, the cost is $128 per

TABLE 2. RESULTS OF CONDENSER MEASUREMENT OF THE STEAM FROM THE 42 X 60 M'INTOSH & HEMPHILL REVERSING ENGINE ON THE 35-INCH BLOOMING MILL OF THE INTERNATIONAL HARVESTER COMPANY AT SOUTH CHICAGO

OBSERVED AND DERIVED VALUES.	UNIT.	TEST NO. 1.	TEST NO. 2.	TEST NO. 3.	TEST NO. 4.
Date of test	1907	Mar. 18	Mar. 18	Mar. 19	Mar. 19
Start of test		12:45 P.M.	3:20 P.M.	9:40 A.M.	3:00 P.M.
Finish of test		1:40 P.M.	4:50 P.M.	11:00 A.M.	4:00 P.M.
Duration of test	Min.	55	90	80	60
CONDENSER.					
Av'ge Initial temp. of cond. water	° Fahr.	43.6	47.0	46.6	47.0
Av'ge Final temp. of cond. water	° Fahr.	139.6	132.1	183.6	174.2
Av'ge Rise of temp. of cond. water	° Fahr.	96.0	85.1	137.0	127.2
Av'ge Head on Venturi meter, Hg	In.	4.36	4.52	2.06	2.3
Av'ge Velocity in Venturi meter throat per second	Feet.	17.88	18.20	12.30	13.0
Av'ge Condensing water per min.	Cu.Ft.	146.1	148.7	100.5	106.2
Av'ge Condensing water per min.	Lbs.	9,110.0	9,280.0	6,260.0	6,620.0
REGENERATOR.					
Av'ge Pressure in regenerator, gage	Lbs.	0.2	0.2	0.32	0.3
Av'ge Steam to regenerator, per min.	Lbs.	835.0	835.0	904.0	921.0
Av'ge Steam to regenerator, per hr.	Lbs.	50,100.0	50,100.0	54,240.0	55,260.0
Av'ge Steam available from auxiliaries, per hour	Lbs.	6,000.0	6,000.0	6,000.0	6,000.0
Av'ge Steam available to turbine per hr.	Lbs.	56,100.0	56,100.0	60,240.0	61,260.0
ENGINE.					
Av'ge I.H.P. during a "mill cycle"	I.H.P.	1,010.0	1,010.0	1,010.0	1,010.0
Av'ge I.H.P. distributed over the hour.	I.H.P.	820.0	820.0	820.0	820.0
Av'ge Steam per ingot by cards	Lbs.	2,090.0	2,090.0	2,090.0	2,090.0
Av'ge Steam per ingot by condenser test	Lbs.	3,100.0	3,010.0	3,140.0	2,910.0
Av'ge Excess to be allowed to steam by cards	Per Cent.	48.4	48.4	50.2	39.3
Av'ge Steam per I.H.P. during "mill cycle" per hour	Lbs.	49.6	49.6	53.7	54.8
Av'ge Steam per I.H.P. distributed over the hour	Lbs.	61.2	61.2	64.2	67.5
MILL.					
Number of ingots rolled during test		15.25	25.0	23.0	19.0
Rate of rolling ingots per hour		16.60	16.7	17.4	19.0
Steam per long ton of 3⅞ x 3⅞ inch billets.	Lbs.	1,200.0	1,230.0	1,280.0	1,190.0
Time for rolling ingots, "mill cycle"	Sec.	217.0	216.0	207.0	189.0

TABLE 1. RESULTS OF TESTS OF 500-KILOWATT INSTALLATION OF RATEAU TURBINE, STEAM REGENERATOR AND GENERATORS, AT THE SOUTH CHICAGO PLANT OF THE INTERNATIONAL HARVESTER COMPANY

OBSERVED AND DERIVED VALUES.	UNIT.	TEST NO. 1.	TEST NO. 2.	TEST NO. 3.	TEST NO. 4.
Date of trial	1907	Mar. 11	Mar. 11	Mar. 12	Mar. 12
Duration of trial	Hrs.	1.00	2.00	2.00	2.00
CONDENSER.					
Av'ge Head on Venturi meter	In.	3.48	7.57	14.72	17.77
Av'ge Initial temp. of condensing water	° Fahr.	38.40	39.30	38.90	39.55
Av'ge Final temp. of condensing water	° Fahr.	80.00	68.30	61.50	66.55
Av'ge Rise of temp. of condensing water	° Fahr.	41.60	29.00	22.60	27.00
Av'ge Condensing water per min.	Cu.Ft.	151.9	192.8	247.2	295.4
Av'ge Condensing water per min.	Lbs.	8,155.0	12,030.0	16,750.0	18,700.0
Av'ge Barometer	In.	29.6	29.6	29.2	29.2
REGENERATOR.					
Av'ge Pressure at turbine, abs.	Lbs.	16.9	16.6	15.9	15.3
Av'ge Temp. of steam at turbine.	° Fahr.	215.5	217.0	216.3	213.2
Av'ge Temp. of air	° Fahr.	48.0	48.0	64.0	64.0
Av'ge Steam delivered per hour	Lbs.	19,500.0	20,220.0	22,050.0	29,280.0
TURBINE.					
Av'ge Pressure above controlling valve	In.	32.9	32.4	30.93	29.75
Av'ge Pressure under controlling valve	In.	18.6	19.0	21.47	24.85
Av'ge Vacuum at exhaust casing	In.	25.31	26.6	26.95	26.40
Av'ge Revs. of turbine + 15 per min.	R.P.M.	1,540.0	1,540.0	1,560.0	1,500.0
Av'ge Brake H.P. at turbine shaft	H.P.	409.0	544.0	727.0	869.0
GENERATORS.					
Av'ge Ammeter readings	Amps.	1,085.0	1,485.0	2,195.0	2,482.0
Av'ge Voltmeter readings	Volts.	245.2	246.9	233.5	238.3
Av'ge Current delivery in kilowatts	Kl.	266.1	366.7	489.2	591.6
Av'ge Efficiency of generators	Per Cent.	90.2	90.2	90.2	90.2
Av'ge H.P. at the switchboard	H.P.	369.0	491.0	656.0	784.0
DERIVED RESULTS.					
Radiation from piping per min.	B.t.u.	2,700.0	2,700.0	810.0	810.0
Heat equiv. of work done per min.	B.t.u.	17,490.0	23,250.0	31,100.0	37,300.0
Heat delivered to condens'g water per min.	B.t.u.	339,100.0	349,000.0	395,100.0	498,100.0
Steam per min. to turbine	Lbs.	326.0	337.0	367.5	488.0
Steam per kilowatt-hour	Lbs.	73.3	55.2	45.2	49.5
Steam per H.P. at switchboard per hr.	Lbs.	52.8	41.2	33.6	37.3
Steam per B.H.P. at turbine per hr.	Lbs.	47.7	37.1	30.7	33.7

TABLE 3. INDICATOR TEST OF ENGINE IN BLOOMING MILL AT THE INTERNATIONAL HARVESTER COMPANY'S PLANT, MARCH 18, 1907.

Twin simple reversing, 42 and 42 x 60 inches. Gear ratio: $\frac{25 \text{ Teeth Engine}}{25 \text{ Teeth Rolls}} = 0.862$. Rolls: Two-high, 35-inch center pinions. Weight of ingot, 2.45 long tons; length, 5.66 feet. Size of product, 4 x 4-inch billets. Time per ingot by test, 190 seconds (from beginning of first pass to end of last pass). Number by test, 18.9; average for 24 hours, 16; maximum for one hour, 19. Revolutions per ingot: Engine, 164; rolls, 141.

	MAX. LOAD.	MAX. I. H. P.	AVER. LOAD.	FRICTION LOAD.
Revolutions per Minute	10.9	100.	51.8	
Mean Effective Pressure	101.	72.15	25.25	4.84
Foot-pounds per Stroke	688,315.	491,900.	172,100.	33,000.
Indicated Horse-power	910.	5,959.	1,080.	

PASS NO.	NO. OF OPENINGS IN ROLLS.	DISTANCE BETW. ROLLS.	ESTIMATED MEAN LATERAL SPREAD.	DIMENSIONS OF PASS.	AREA IN SQ. IN.	REDUCTION OF AREA. Square Inch.	REDUCTION OF AREA. Per Cent.	LENGTH AFTER PASS, Feet.	ELONGATION. Feet.	ELONGATION. Per Cent.	ROLLS. REVOLUTIONS. Diameter, In.	REVOLUTIONS. Pass.	REVOLUTIONS. Interval.	REVOLUTIONS. Pass + Interval.	TIME IN SECONDS. Pass.	TIME IN SECONDS. Interval.	TIME IN SECONDS. Pass + Interval.	Revolutions of Engine Shaft.	M. E. P. Pass + Interval. Right Head.	M. E. P. Pass + Interval. Left Crank.	M. E. P. Pass + Interval. Total.	M. E. P. Pass + Interval. 2 Times Total.	TOTAL INDICATED WORK. Pass + Interval in Foot-pounds.	FRICTION LOSS. At 132,000 Ft.-lb. per Revolution.	NET INDICATED WORK. Pass.	NET INDICATED WORK. Per Foot Length.	STEAM CONSUMPTION. TOTAL POUNDS. Right Head.	Left Crank.	Total.	2 Times Total.	Rate per I. H. P. Hour.	PASS NO.
1	1	18½	¼	18½ × 17½	319.1	3.9	1.2	5.73	0.07	1.2	27	0.8	2.0	2.8	2.3	2.8	5.1	3.3	102.6	122.1	224.7	449.4	3,062,000	436,000	2,626,000	458,000	10.64	11.89	22.53	45.06	29.14	1
2	1	17	¼	17 × 17½	297.5	21.6	6.8	6.15	0.42	7.3	27	0.9	2.7	3.6	3.0	3.5	6.5	4.2	107.1	90.1	197.2	394.4	2,687,000	554,000	2,133,000	347,000	11.53	9.48	21.01	42.02	30.97	2
3	TURN 1	15½	¼	15½ × 17½	267.4	30.1	10.1	6.84	0.69	11.2	27	1.0	2.0	3.0	3.8	3.4	7.2	3.5	112.3	112.0	224.3	448.6	3,059,000	462,000	2,597,000	380,000	13.08	10.88	23.96	47.92	31.04	3
4	1	13	¼	13 × 17½	227.5	39.9	14.9	8.04	1.20	17.5	27	1.1	2.4	3.5	5.7	4.2	9.9	4.1	122.4	98.9	221.3	442.6	3,018,000	541,000	2,477,000	308,000	14.81	10.50	25.31	50.62	33.20	4
5	TURN 2	15	¼	15 × 13½	198.8	28.7	12 6	9.21	1.17	14.6	25	1.4	0.8	2.2	4.0	1.4	5.4	2.5	199.5	164.3	363.8	727.6	4,961,000	330,000	4,631,000	503,000	18.91	17.66	36 57	73.14	29.19	5
6	2	13	¼	13 × 13½	175.5	23.3	11.7	10.43	1.20	13.2	25	1.6	3.1	4.7	2.5	4.4	6.9	5.4	107.9	150.6	258.5	517.0	3,523,000	713,000	2,890,000	270,000	11.08	16.23	27 31	54.62	30.70	6
7	TURN 2	11¾	¼	11¾ × 13¼	155.7	19.8	11.3	11.75	1.32	12.7	25	1.8	2.0	3.8	2.3	2.9	5.2	4.4	169.7	160.5	330.2	660 4	4,498,000	581,000	3,917,000	333,000	17.79	17.28	35.07	70.14	30.87	7
8	2	10½	¼	10½ × 13½	141.8	13.9	8.9	12.91	1.16	9.9	25	2.0	1.7	3.7	2.9	2.5	5.4	4.3	120.7	118.8	239.5	479.0	3,265,000	568,000	2,697,000	209,000	15.09	13 34	28.43	56.86	34.48	8
9	2	9¼	0	9¼ × 13½	124.9	16.9	11.9	14.65	1.74	13.5	25	2.2	1.2	3.4	3 3	2.8	6.1	3.9	124.4	130.5	254.9	509.8	3,475,000	515,000	2,960,000	202,000	15.76	14.97	30.73	61.46	34.03	9
10	2	8	0	8 × 13½	108.0	16.9	13.5	16.95	2.30	15.7	25	2.6	2.2	4.8	3.0	4.6	7.6	5.6	113.7	179.5	293.2	586.4	3,992,000	739,000	3,253,000	292,000	12.78	17.49	30.27	60.54	30.00	10
11	TURN 3	10½	¼	10½ × 8½	86.6	21.4	19.8	21.12	4.17	24.6	26	3.1	1.1	4.2	4.2	3.9	8.1	4.9	226.0	226 4	452.4	904.8	6,162,000	647,000	5,515,000	261,000	24.57	23.67	48.24	96.48	31.00	11
12	3	8	¼	8 × 8½	68.0	18.6	21.5	26.91	5.79	27.4	26	4 0	1.6	5.6	4.6	4.7	9.3	6.5	286.3	270.6	556.9	1,113.8	7,585,000	858,000	6,727,000	250,000	29.32	28.49	57.81	115 62	30.16	12
13	TURN 3	7	¼	7 × 8¼	57.8	10.2	15.0	31.65	4.74	17.6	26	4.7	0.7	5.4	4.8	2.6	7.4	6.3	231.3	170.1	401.4	802.8	5,458,000	832,000	4,626,000	146,000	28.51	23.38	51.89	103.78	37.65	13
14	3	6	⅛	6 × 8½	51.0	6.8	11.8	35.87	4.22	13.3	26	5.3	1.9	7.2	4.6	4.1	8.7	8.3	208.1	162 9	371.0	742.0	5,057,000	1,096,000	3,961,000	110,000	27.74	20.87	48.61	97.22	38.06	14
15	TURN 4	7	⅛	7 × 6½	42.9	8.1	15.9	42.65	6.78	18.9	27	6.0	1.1	7.1	5.7	2.9	7.7	8.2	217.3	247.1	464.4	928.8	6,321,000	1,082,000	5,239,000	123,000	28.12	29.06	57.18	114.36	35.80	15
16	4	6	⅛	6 × 6¼	37.5	5.4	12.6	48.78	6.13	14.4	27	6.9	1.7	8.6	6.4	8.8	15.2	10.5	162.3	167.1	329.4	658.8	4,490,000	1,386,000	3,104,000	64,000	24.07	24.25	48.32	96.64	42.63	16
17	TURN 4	5	⅛	5 × 6⅛	30.6	6.9	18.4	59.80	11.02	22.6	27	8.5	1.1	9.6	6.5	3 4	9.9	11.1	327.0	283.3	610.3	1,220.6	8,321,000	1,465,000	6,856,000	115,000	36.25	38.49	74.74	149.48	35.56	17
18	4	3⅞	⅛	3⅞ × 6¼	24.2	6.4	20.9	75.60	15.80	26.4	27	10.7	1.5	12.2	8.3	5.5	13.8	14.1	356.8	319.0	675.8	1,351.6	9,217,000	1,861,000	7,356,000	97,000	45.20	41.44	86.64	173.28	37.25	18
19	TURN 5	5	⅛	5 × 4	20.0	4.2	17.4	91.50	15.90	21.0	27	12.9	0.2	13.1	11.0	3.1	14.1	15.2	374.2	384.2	758.4	1,516.8	10,338,000	2,006,000	8,332,000	91,000	48.38	51.04	99.42	198.84	38.12	19
20	5	3⅞	¼	3⅞ × 4¼	16.5	3.5	17.5	110.90	19.40	21.2	27	15.7	0.9	16.6	13.0	4.5	17.5	19.2	337.4	346.2	683 6	1,367.2	9,317,000	2,534,000	6,783,000	61,000	50.80	60.95	111.75	223.50	47.50	20
21	TURN 5	4	⅛	4 × 4	16.0	0.5	3.0	114.40	3.50	3.2	27	16.2	....	16.2	13.0	....	13.0	18.8	182.4	192.3	374.7	749.4	5,107,000	2,482,000	2,625,000	23,000	35.33	44.18	79.51	159.02	61.65	21
				Total.....								109.4	31.9	141.3	114.9	75.1	190.0	164.3	4189.4	4096.5	8285.9	16,571.8	112,913,000	21,688,000	91,225,000		519.76	525.54	1045.30	2090.60	36.65	
				Per ton product.....																			46,087,000.....							853.30		

indicated horse-power, delivered. With the turbine in operation the cost per horse-power per annum is reduced to $45.70, or a saving to the mill of $67,500 on the basis of 820 indicated horse-power from the roll-train engine.

Fig. 1 exhibits various observed and derived quantities concerning a "mill cycle." Information of this character is seldom available to the mill manager, and is offered as a partial illumination of a subject which has recently exacted much labor and attention. The upper diagram shows the actual energy delivered at the roll surface, and is to be rendered compatible with the diagram, showing mean effective pressures, immediately underneath, by asserting that the tangential effort at the roll surface is very nearly constant on a straight bar, while the inertia effect and crank positions cause the variations of mean effective pressure.

Tachometer records follow below as the specific guide to the estimation of the power required for the several passes. All of the important information derivable from the roll-train observations is exhibited in Table 3.

Fig. 4 is an attempt to show on a volume-pressure diagram the relations of the work done by the engine, and that given out by the turbine, for a given quantity of steam passing through the system. Taking account of all losses, and assuming that all of the steam from a reversing engine is passed through turbines of sufficient capacity, the power of the engine would be represented by 820, and of the turbines by 1510. The combined diagram under the curve inclosing an area representing the total available energy from the steam would have areas proportionate to the above numbers. It will be seen that the steam enters the turbine blades, first stage at an absolute pressure of 13.2 pounds, and is expanded through the successive stages to an absolute pressure of 1.5 pounds, or approximately nine expansions. It is utterly impossible to maintain this low back pressure on the low-pressure piston of any compound mill engine of the reversing type, or to undertake one-third of this number of expansions.

The low-pressure turbine occupies a territory for the conversion of heat into work that cannot be appropriated by any other form of heat motor. The practical conclusion is warranted, therefore, that the combination of the ordinary reversing engine with a low-pressure turbine will return more power for a given weight of steam than any system of compounding of such engines.

Assuming that the compound reversing mill engine is large enough to operate the rolls as a non-condensing engine, it is easily seen that it is a means of great saving to pass the steam direct to a steam turbine. Generally the size of reversing mill engine is sufficient to meet this requirement, in view of the fact that the pressure losses at the throttle are notoriously large, as such engines usually operate.

The foregoing remarks apply with greater force to any ordinary reversing mill engine. Sooner or later the application of low-pressure steam turbines will prove the advisability of running mill engines with the highest practicable back pressure that wide-open throttles and the highest initial steam pressures will allow.

The Gas-power Situation in America

By F. E. Junge

Since I had the pleasure of presenting to American engineers a series of papers and reports on the "Design, Construction and Application of Large Gas Engines in Europe"—which were the result of a literary campaign undertaken for Power during the year 1905—and since my contributions to the understanding of this subject have been received in a very friendly spirit, I venture to think that a discussion of the gas-power situation in the United States from the viewpoint of a foreign observer of American industrial conditions may be of value in throwing additional light on some features of a more commercial than technical order, which do not stand out clear enough when reviewed from within the national horizon.

A short résumé of the growth and extent of power application in the United States will at once reveal the extreme importance which the discussion and treatment of this factor possesses in the field of industrial economics, and will serve at the same time to emphasize the general modern tendency of profit-making management, which may be condensed in the following categoric demand: Seize this time as the very best for introducing such sure and lasting economics as will make their possessors sure of a competence and a profitable business, however lean may be some of the years to come.

According to the report from the United States census the total power employed in this country in 1870 was 2,346,000 horse-power; in ten years it had increased 45 per cent., that is, to 3,411,000 horse-power; by 1890 it had advanced 75 per cent., to 5,955,000, and by the end of the century it was 10,410,000, a further increase of 75 per cent. Last year the aggregate was 14,465,000 horse-power, or an advance of 39 per cent. Should the rise continue during the next few years at the same rate, then the increase for the present decade will be no less than 93 per cent.

Before 1890 the main sources of power were steam and water, but since that time gas and electric power have made enormous strides, the advance from 1890 to 1900 amounting to 1400 per cent. in the case of gas, and very nearly 2000 per cent. in the case of electric power. Electric power totals now 1,138,000 horse-power, the remainder being divided between gas and miscellaneous power. It should be remembered, however, that a considerable part of electric power is converted from gas power, that is, delivered from gas-engine-driven generators. It is interesting to note that about 35 years ago water power represented nearly one-half of the total, while now it represents only 11.5 per cent.

The total gas-engine horse-power employed was 289,514 in 1905, 134,742 in 1900 and 8930 in 1890. The percentage of increase from 1900 to 1905 was 114.9 per cent., only second, as regards increase, to water power; but, of course, steam leads by a long head. The returns for 1905 show 10,664,560 horse-power.

In order to appreciate the economic importance of these figures it is necessary to compare them with the corresponding amount of coal that must be expended from year to year for their realization. The world's total annual coal production and consumption has increased since the time of the invention of the steam engine, that is, within one hundred and twenty-five years, to about fifty times its original value. The total amount of coal consumed in United States manufactures runs up now to almost 100,000,000 tons per year, representing about 25 per cent. of the total annual coal production. Figuring at an average price of $2 per ton, it is obvious that the universal adoption of a method of generation which will render an equivalent power service with one-half the consumption will represent an annual saving in treasure of some $100,000,000, disregarding entirely what the saving in fuel means to future generations.

With these facts in mind I propose to discuss shortly, first, the literary treatment which the subject of gas-power engineering is receiving in best books, periodicals and professional papers, such as are edited by the various engineering societies; secondly, the external obstacles offered to progressive engineering by imparting undue prominence to the selling end of manufacture, and thirdly, the internal difficulties militating against a healthy industrial growth of the gas-power branch, by exaggerating the importance of quantity and immediacy of earnings at the expense of profitable stability.

So far as theoretical and practical research have been able to analyze the very complex process of converting combustible matter into useful power directly and without the intervention of inefficient and passive apparatus, and so far as the technical literature of this country can be regarded as reflecting the light of these investigations, it appears that the few scientific treatises available on the sub-

ject have been devoted to one of the three aims: The abstract study of the internal thermo-dynamic relations of gas-producing and converting apparatus; the quantitative side of design of their mechanisms, and the historical or inventive evolution of the problem. All of these modes of treatment are interesting in themselves and valuable in their combination. Perceptibly the cultivation of the mechanical engineering part, that is, the theory of the construction of gas producers and engines, as an expedient in the practical development in the art is most vital to the question, greatly more so than is a purely hypothetical analysis of the physical and chemical phenomena occurring in the working cycle.

There is, however, one more side to the subject, which so far has been entirely neglected and the want of which is acutely felt by the trade, namely, the broad economic aspect of the gas-power problem, its relation to the iron, coal and kindred industries, and its scientific and impartial presentation to men, professional and mercantile. There should be more treatises to appeal to the interests of wider industrial and commercial circles, demonstrating the possibilities and clarifying the limitations of the application of gas power in the varied industries.

To explain why Europe, and especially Germany, has made such remarkable strides in the utilization and conservation of fuels is easier for the foreigner than it is for him to analyze the obstacles which have been and are yet militating against a stable and healthy industrial growth of gas power in this country. It is owing to the scarcity of natural resources, to the density of population, to the concentration of industries, and to the keenness of competition, that the law of self-preservation has impressed itself with more weight on the responsible quarters of old Europe, and that the demand to economize by reducing waste in methods of production has forced itself earlier upon the German engineer, than on any other members of the profession. And if I say that the records which have been presented in POWER refer to achievements in a branch of engineering which has reached a more advanced phase of development than any of its European competitors, it is only after years of comparative study that I feel justified in stating it so boldly.

Confining my remarks strictly to the consideration of power as a potential factor in the industrial arts, I submit the following criticism of the situation in America, as I see it.

The scarcity of good literature available on gas-power engineering in this country is due, in the first place, to the unfortunate but incontrovertible fact that those men whose learning and experience would enable them adequately to discuss and propagate the subject in professional circles are not permitted by the firms controlling their services to make public such data as would be of direct practical usefulness to the working engineer. To the foreign observer of American industrial methods this policy of reserve unquestionably appears distinct from the broad-minded course of action which is pursued by the majority of other governmental or private institutions. In fact, it seems dictated by traditional rather than by logical motives, since there certainly would be less injury done to the commercial interests of manufacturers by a liberal exchange of useful information than by promoting—as they deem wise to—the publication of literature of the sort as is nowadays distributed by their publicity men. Whether or not this hermit-like attitude of isolation and secrecy, as Dr. Lucke terms it, be beneficial to the earning end of manufacture, there can be no doubt that it works direct harm to the propagation of technical truth among professional men. It is owing to this very secrecy that the technical student of this branch of engineering cannot, except for the merely fundamental principles involved, rely on what form otherwise and elsewhere the most valuable sources of information, namely, the current technical periodicals, and the professional papers read before the national engineering societies. For the majority of material and discussion advanced, so far as it is not of an entirely academic nature, relates to efficiencies and performances of commercial machinery, being obtained from the advertising or publicity staffs of the large manufacturing concerns, who, no doubt, withhold all information except that which presents the good features, and those only, of their products.

I am bold enough to go so far as to say that the commercial success of any novel machine depends, in this country, more on the ability of the sales department than on the real technical merits of its design. For it is conceded that fuel cost, which is still considered by some as a secondary factor in power-plant economics, is hardly ever ascertained, after a period of probation in practice, by impartial investigators, since manufacturers and users alike are yet pleased to get continuous and reliable service at almost any price. Satisfaction such as this, though comfortable for the builder, is far from conducive to further progress in the mechanic arts. In fact, it is prejudicial to it.

Another obstacle, and one equally traceable to other lines of manufacture, is that after a few years of unchallenged growth too much capital has been invested in one special form or pattern, and not enough of skilled talent has been provided to allow of any radical changes to be instigated. Profit-making management, so long as it is concerned with present dealings only, must disapprove of the frequent alterations in design so essential in a developing field, because investors clamor for immediate returns, and for the greatest possible revenues from the capital invested.

Progressive engineering in general, and our special branch in particular, would ultimately derive much greater benefit from careful experimental investigation and thorough scientific research, if the key-note of opportunism and immediacy were not so preponderatingly sounded: Immediate results, immediate satisfaction, immediate earnings! The full measure of prosperity in the fat years, and the ability to tide over the lean ones, are the results of forethought and preparation of a different character. The stability of modern industry requires that some money, some labor, and some time be expended for operations other than those which tend to produce the maximum income immediately.

But here is where this special trouble is wedded to the lamentable plague of commercialized engineering which is threatening us today in all progressive countries: Monopoly, the great discourager of effort in trade, manufacture and profession, will dictate the technical methods to be followed, thereby suppressing individual aspirations; few great corporations will buy up the highest grade of skilled talent, literary, inventive, mechanical, but will keep it latent or, at least, barren of creative work, by raising almost insuperable barriers to the propagation of the fruits of its labor.

Gas-power engineering is no exception to the rule. It is in spite of the antagonistic attitudes exhibited by the large manufacturing corporations, and without their sanction that the writer has begun his campaign of publicity, moved by the earnest desire to reciprocate for the wealth of impressions received during his studies in this country, in setting before the ambitious and studious reader such achievements, and the ways for attaining them, as have been realized in this line abroad. Happily for the development of the art and for the purse of the power user, the destiny of this particular industry is not controlled yet, as is that of the electrical industry, by a few huge co-operating corporations, and an alternative to unsatisfactory dealing is still available. It is hoped that the vastness and multiplicity of interests involved in the gas-power field will effectively prevent centralization of the kind that precludes a stable and healthy industrial growth.

The Stevens Institute of Technology has appointed Miss E. M. Hawkins, graduate of the Pratt Institute Library School, to take charge of its library. An important feature of the library is the section devoted to patent literature, containing issues of the *Official Gazette* and specifications of the United States Patent Office. Prof. W. H. Bristol, well known in the field of invention, has contributed liberally to the support of this section.

Fans for Draft, Forced and Induced—II

Directions for Calculating Proportions and Capacity of Fans, the Horse-power Required, and the Volume of Air Delivered

BY CHARLES L. HUBBARD

Horse-power Required

The horse-power required for moving air through a short pipe (neglecting friction) is given by the equation

$$H.P. = \frac{P \times A \times V}{550}, \qquad (a)$$

where P = pressure per square foot producing the flow.
A = area of pipe in square feet.
and V = velocity of flow in feet per second.

Now $\frac{P}{5.2} = h$, in which h is the equivalent pressure in inches of water; and from this we may write $P = 5.2 \times h$.

$A \times V$ = cubic feet of air discharged per second = Q'.

Substituting these values in equation (a) gives

$$H.P. = \frac{5.2 \times h \times Q'}{550}. \qquad (b)$$

Going back to the formula for the relation between velocity and pressure, we have

$$v = 66\sqrt{h}.$$

$$\frac{v}{66} = \sqrt{h}.$$

$$\frac{v^2}{4356} = h.$$

Substituting this value of h in equation (b) we have

$$H.P. = \frac{5.2 \times v^2 \times Q'}{4356 \times 550} = v^2 \times Q' \times 0.0000220 \qquad (c)$$

This formula gives simply the power required for moving the air, and does not take into account the friction of the fan itself.

In order to make it available for practical use, the factor for efficiency must be introduced, and the equation finally becomes:

$$H.P. = \frac{0.0000022\ v^2\ Q'}{E} \qquad (d)$$

where v = velocity of air flow through the fan outlet = peripheral velocity in feet per second × 0.65.
Q' = cubic feet of air delivered per second.
and E = efficiency of the fan for the given speed.

The efficiency of a fan varies with the peripheral velocity and the percentage of free outlet.

The efficiency at various speeds will depend somewhat upon the type and make of fan, but for average conditions with a free outlet, Table IV may be used. For speeds between those given in the table, the factor corresponding to that nearest to the required speed may be used, or factors may be interpolated if desired.

TABLE IV.

DOUBLE INLET FANS, WITH FREE DISCHARGE.

Peripheral Velocity, in Feet per Minute.	Efficiency.
1000	0.05
1500	0.08
2000	0.10
2500	0.12
3000	0.14
3500	0.16
4000	0.18
4500	0.20
5000	0.22
5500	0.24
6000	0.26
6500	0.28

The methods of determining the size and speed of fan and the necessary power for driving it under actual conditions can best be illustrated by working a practical

Example

What will be the necessary size and speed of fan, and power of engine for furnishing 10,000 cubic feet of air per minute, under a static pressure of 1 inch of water? The system is to be of the forced-draft type with the fan located near the boilers.

In working problems of this kind it is necessary to first assume a velocity of air flow through the fan outlet, and this under average conditions may be taken as about 2000 feet per minute.

If in the above case the volume is 10,000 cubic feet, the required area of outlet will be 10,000 ÷ 2000 = 5 square feet, which corresponds to a 4-foot fan.

The next step is to find the necessary speed to maintain the required pressure when delivering this quantity of air. It has been shown that the air velocity pressure is equal to the dynamic pressure minus the static pressure, or in other words, the dynamic pressure is equal to the static pressure plus the air velocity pressure, and may be expressed as follows:

$$DP = SP + AVP$$

In the present example, $SP = 1$, and $AVP = 0.25$, the latter being computed for a velocity of 2000 feet per *minute* by the formula $v = 66\sqrt{h}$. From this, $DP = 1 + 0.25 = 1.25$, and the ratio

$$\frac{AVP}{DP} = \frac{0\ 25}{1\ 25} = 0\ 20$$

Referring back to Fig. 6 and following down the curve marked $\frac{AVP}{DP}$, we find for a ratio of 0.20 the corresponding ratio of air velocity pressure to peripheral velocity pressure $\frac{AVP}{PVP}$ as given by the other curve to be approximately 0.18, and the percentage of opening 0.64.

If the ratio of air velocity *pressure* to peripheral velocity *pressure* is 0.28, the ratio of the corresponding *velocities* will be equal to $\sqrt{0.18} = 0.43$, and the required peripheral velocity will be 2000 ÷ 0.43 = 4650 feet per minute. The circumference of a 4-foot fan is 12.5 feet; therefore the required speed will be 4650 ÷ 12.5 = 322 revolutions per minute.

The horse-power is first computed for a fan of this size and speed discharging into free air, and then reduced in proportion to the equivalent reduction in outlet, which in this case is 0.64.

Tests show that when the outlet is restricted, the static and dynamic pressures increase, and the air velocity pressure decreases, thus reducing the volume of air moved; at the same time, the efficiency increases until the area is reduced to about forty or fifty per cent. of the full opening.

These various changes in factors affecting the horse-power make it necessary, theoretically, to compute the power separately for each variation in resistance or pressure.

This, however, is much simplified in actual practice because it so happens that for reductions in outlet area down to about fifty per cent., the power varies almost in direct ratio with the volume of air moved, that is, with the reduction of outlet area.

Applying this rule to the present case, we have

$$H.P. = \frac{0.0000022 \times v^2 \times Q' \times 0.64}{E},$$

$v = \frac{4650}{60} \times 0.65 = 50$, from which $v^2 = 2500$.

$Q' = 50 \times 5 = 250$ cubic feet.

$E = 0.20$ (from Table V).

Substituting these values in the above formula, we have

$$H.P. = \frac{0.0000022 \times 2500 \times 250 \times 0.64}{0.20} = 4.4$$

Table V has been computed in this manner for pressures ranging from 0.5 to 2.5 inches of water column for different sizes of fans based on an air velocity of 2000 feet per minute through the outlet.

TABLE V.

FOR FORCED DRAFT, TEMP. OF AIR 60°.

Diameter of Fan.	Cubic Feet of Air Delivered to Furnace per Minute.	Pressure in Inches of Water. 0.5 R.P.M.	0.5 H.P.	0.75 R.P.M.	0.75 H.P.	1.00 R.P.M.	1.00 H.P.	1.25 R.P.M.	1.25 H.P.	1.50 R.P.M.	1.50 H.P.	2.00 R.P.M.	2.00 H.P.	2.50 R.P.M.	2.50 H.P.
2' 6"	4,200	510	1.6	560	1.8	600	1.9	640	2.1	710	2.3	780	2.5	850	2.7
3'	5,800	430	2.2	460	2.4	490	2.6	530	2.8	590	3.1	640	3.4	710	3.8
3' 6"	7,800	360	3.0	400	3.3	420	3.5	450	3.8	500	4.2	550	4.6	610	5.1
4'	10,000	320	3.9	350	4.2	370	4.4	400	4.9	440	5.4	480	5.9	530	6.5
4' 6"	12,400	290	4.8	310	5.2	330	5.6	360	6.0	400	6.7	430	7.3	470	8.0
5'	15,200	250	5.9	270	6.4	290	6.8	310	7.4	350	8.2	380	8.9	420	9.8
5' 6"	18,200	230	7.0	250	7.7	270	8.2	300	8.8	330	9.8	360	10.6	390	11.8
6'	21,400	210	8.3	230	9.1	250	9.6	260	10.4	290	11.5	320	12.5	350	13.9
7'	28,800	180	11.2	200	12.2	210	13.0	230	14.0	250	15.5	280	16.8	300	18.7
8'	37,200	160	14.4	170	15.7	190	16.7	200	18.1	220	20.1	240	21.8	270	22.5
9'	46,800	140	18.1	160	19.8	170	21.1	180	22.7	200	25.3	220	27.4	240	30.3
10'	57,400	130	22.2	140	24.3	150	25.8	160	27.9	180	3.1	200	33.6	210	37.2

All previous calculations in connection with fans have been based on an air temperature of 60 degrees.

In the case of induced draft, where the gases reach the fan at a temperature of 400 to 600 degrees, certain corrections must be made because of the increase in volume of a given weight, due to the higher temperature.

The principles upon which the necessary corrections are made are as follows:

1. When air is heated and expands, the weight of a given volume varies as the absolute temperature.

2. The necessary fan speed to produce a given pressure is proportional to the square root of the absolute temperature.

3. The power required to drive a fan varies as the velocity of air flow, when the pressure and area of outlet remain constant.

The factors given in Table VI have been computed for a rise in temperature from 60 to 400, 500 and 600 degrees, and may be used for correcting the air, quantity, speed, and horse-power of fans used for induced draft.

TABLE VI.

Temperature of Gases. Degrees F.	Factor for Proportional Volume at 60 Degrees.	Factor for Increase in Speed.	Factor for Increase in Horse-power.
400	0.77	1.28	1.28
500	0.73	1.35	1.35
600	0.70	1.42	1.42

EXAMPLE: A 3-foot fan running at a speed of 590 revolutions per minute discharges 5800 cubic feet of air per minute, at a temperature of 60 degrees, against a pressure of 1.5 inches of water, and requires 3.1 horse-power. What speed and horse-power will be required to maintain the same pressure when used for induced draft, with the gases at a temperature of 600 degrees, and what volume of air at a temperature of 60 degrees will be drawn into the furnace?

Volume, $5800 \times 0.70 = 4060$ cubic feet per minute. Speed, $590 \times 1.42 = 838$ revolutions per minute; horse-power, $3.1 \times 1.42 = 4.4$.

In the factors for horse-power, no account has been taken of the small increase in efficiency due to the higher peripheral velocity, as it is on the side of safety.

Table VII has been computed in this manner for the flue-gas temperatures of 500 degrees, which may be taken as the average for induced draft plants where no economizer is used.

Regulation is generally obtained by means of a throttle valve in the steam pipe, which is operated automatically by the pressure in the boiler, thus changing the speed of the fan as may be required to give the proper draft to maintain a constant boiler pressure.

When the required volume of air comes between those given in the Tables V and VII, the speed may be increased or diminished to make up for small differences, but for any considerable variation the width of the fan should be changed, keeping the diameter and speed the same, in order to maintain the pressure, which depends directly upon the peripheral velocity.

The air volumes in these tables are computed on the basis that the width of fan wheel at the inlet is 0.52 of the diameter. This gives the following widths, from which the volume per inch in width can be computed for any given diameter of wheel.

TABLE VIII.

Diameter of Wheel.	Width of Wheel.
2 feet, 6 inches.	16 inches.
3 "	19 "
3 " 6 "	22 "
4 "	25 "
4 " 6 "	28 "
5 "	31 "
5 " 6 "	34 "
6 "	37 "
7 "	43 "
8 "	50 "
9 "	56 "
10 "	62 "

The use of the tables may be illustrated by a few practical examples.

EXAMPLE: It is desired to burn 1800 pounds of coal per hour under a battery of boilers, using a forced draft of 1.25 inches of water column. What size and speed of fan will be required, and what horse-power of engine?

As $1800 \div 60 = 30$ pounds of coal per minute, it requires $30 \times 230 = 6900$ cubic feet of air.

From Table V we find that a 3½-foot fan running at 450 revolutions per minute will deliver 7800 cubic feet of air per minute under a pressure of 1.25 inches of water, and requires 3.8 horse-power. The air volume required is 6900 cubic feet; or $6900 \div 7800 = 0.88$ of that delivered by a standard wheel 22 inches in width, so that under the above conditions the required width of wheel would be $22 \times 0.88 = 20$ inches.

Theoretically, the horse-power varies as the volume of air moved; but actually the friction of the fan forms a considerable part of the load, and this would be affected but little by any slight change in

TABLE VII.

FOR INDUCED DRAFT, TEMP. OF FLUE GASES 500°.

Diameter of Fan.	Cubic Feet of Air at 60° Temp. Drawn into Furnace per Minute.	Pressure in Inches of Water. 0.5 R.P.M.	0.5 H.P.	0.75 R.P.M.	0.75 H.P.	1.00 R.P.M.	1.00 H.P.	1.25 R.P.M.	1.25 H.P.	1.50 R.P.M.	1.50 H.P.	2.00 R.P.M.	2.00 H.P.	2.50 R.P.M.	2.50 H.P.
2' 6"	3,000	688	2.2	756	2.4	810	2.6	864	2.8	958	3.1	1053	3.4	1147	3.6
3'	4,200	580	3.0	621	3.2	661	3.5	715	3.8	796	4.2	864	4.6	958	5.1
3' 6"	5,700	486	4.0	540	4.5	567	4.7	607	5.1	675	5.7	742	6.2	823	6.9
4'	7,300	432	5.3	472	5.7	500	6.1	540	6.6	594	7.3	648	8.0	715	8.8
4' 6"	9,300	390	6.5	418	7.0	445	7.5	486	8.1	540	9.0	580	9.8	634	10.8
5'	11,100	337	8.0	364	8.6	391	9.2	418	10.0	472	11.1	513	12.0	567	13.2
5' 6"	13,300	310	9.5	337	10.4	364	11.1	405	11.9	445	13.2	486	14.3	526	15.9
6'	15,600	283	11.2	310	12.3	337	13.0	351	14.0	391	15.5	432	16.9	472	18.7
7'	21,000	243	15.1	270	16.5	283	17.5	310	18.9	337	20.9	378	22.6	405	25.2
8	27,100	216	19.4	230	21.2	256	22.5	270	24.4	297	27.1	324	29.4	364	30.4
9'	34,200	189	24.4	216	26.7	230	28.5	243	30.6	270	34.1	297	37.0	324	40.9
10'	41,900	175	30 0	190	32.8	202	34.8	216	37.6	243	41.8	270	45.3	283	50.2

the width; so in practice it is best to take the horse-power directly from the table without change.

In the above example a standard 3½-foot fan would probably be used without any reduction in width, and any excess of air would be cared for by automatic regulation of speed.

In the case of large fans this is of more importance, and must be dealt with accordingly.

When induced draft is used, a much larger volume of air must be handled by the fan, due to the high temperature and resulting expansion. The weight, and consequently the volume of air at 60 degrees which is necessary for the combustion of a given weight of fuel is, of course, the same whether forced into the ash-pit or drawn in by suction, but in the latter case, when the heated gases reach the fan the volume is much increased and the density correspondingly decreased.

If a fan supplying forced draft should be used to produce an induced draft of the same intensity, the following changes would take place, their magnitude depending upon the temperature of the gases.

First, the speed would have to be increased in order to maintain the same pressure; second, the power necessary to drive the fan at the higher speed would be greater than before; and third, the volume of air supplied to the furnace would be reduced.

In order to adapt Table V to the conditions found in induced draft, it is necessary to make certain corrections which have already been described; these for the average flue-gas temperature of 500 degrees may be taken as follows:

Volume $\times$ 0.73, Speed $\times$ 1.35, H. P. $\times$ 1.35.

EXAMPLE: A 4-foot fan when moving air at 60 degrees temperature will deliver to the ash-pit of a boiler 10,000 cubic feet per minute and maintain a pressure of 1 inch of water at a speed of 370 revolutions per minute, and requires 4.5 horse-power. What will be the volume of air drawn into the ash-pit by the same fan when used for induced draft, and what speed and horse-power will be required to maintain the same pressure?

Using the factors previously given, we have,

Volume, $10{,}000 \times 0.73 = 7300$ cubic feet,
Speed, $370 \times 1.35 = 500$ revolutions per minute,
Horse-power, $45 \times 1.35 = 6.1$.

When an economizer is used, the temperature of the gases is considerably reduced, which results in a larger air volume for a given fan and a lower speed and less power; but on the other hand, the resistance is increased, so that these effects are offset, and Table VII may be used as it stands for economizers of medium depth and ample area for air flow between the pipes.

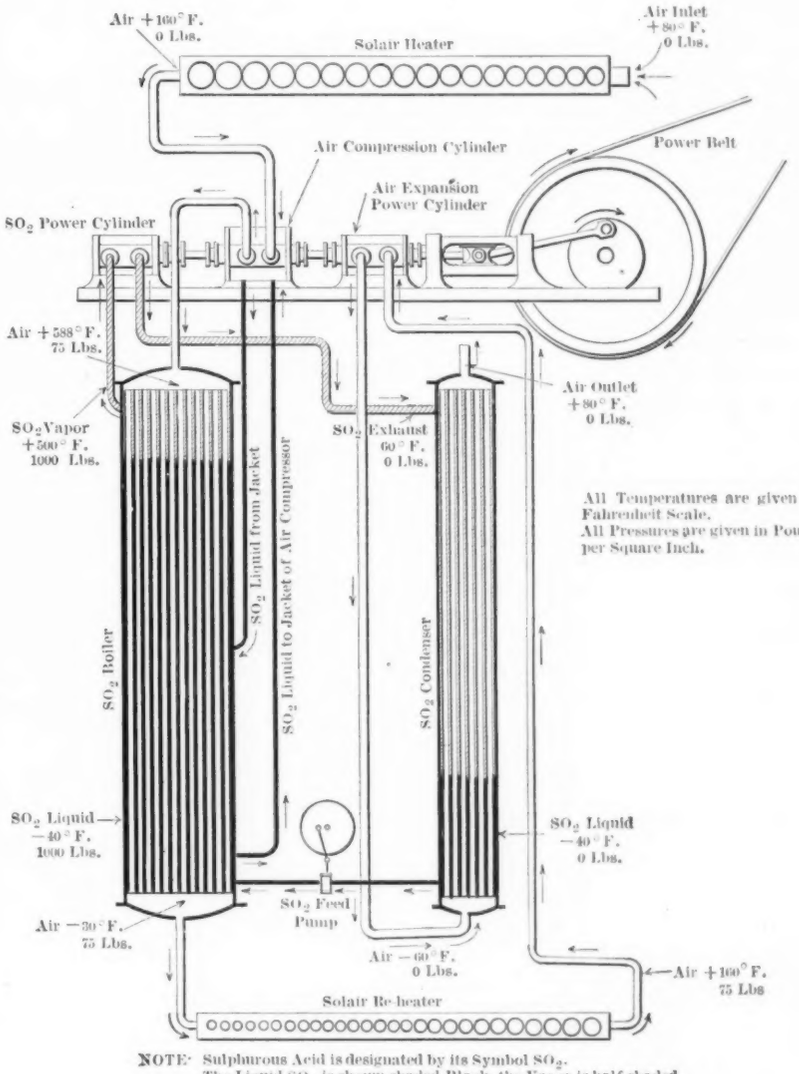
For deep economizers, a larger fan should be used than called for in the table in order to secure the required air volume without increasing the pressure, which would be the case if a smaller fan was used at a higher peripheral velocity.

The use of Table VII is practically the same as that of Table V. The volumes in the second column are for an air supply at a temperature of 60 degrees, computed in the same manner as for forced draft, while corrections have been made for volume, speed, and horse-power.

Builders of mechanical draft apparatus usually furnish an engine of standard size with each fan of sufficient power to drive it at the probable maximum speed under the conditions of each particular case.

An Illusory Hot Air and Sulphur Dioxide Power System

The accompanying diagram illustrates a proposed composite heat engine which is supposed to give a thermal efficiency of



DIAGRAMMATIC REPRESENTATION OF A PROPOSED COMPOSITE HEAT ENGINE

EXAMPLE: Take the case of the previous example, and apply induced draft instead of forced draft.

The required air volume is 6900 cubic feet per minute at a pressure of 1.25 inches of water.

Looking in Table VII we find a 4-foot fan will supply 7300 cubic feet per minute, and will maintain the desired pressure at a speed of 540 revolutions per minute, and requires 6.6 horse-power.

more than 100 per cent. Two power cycles are involved, the medium in one being air and that in the other sulphur dioxide. The air is taken in through a system of pipes inclosed in a glass box, called by the inventor of the system a "solair heater." In this the temperature of the air is supposed to be raised 80 degrees Fahrenheit. It passes then to a compressor which increases the pressure from atmospheric to 75 pounds (90

pounds absolute) and consequently raises the temperature from 160 to 588 degrees. The compressor delivers this heated air to a sulphur dioxide boiler, where it heats the sulphur dioxide from 40 degrees below zero to 500 degrees above. The sulphur dioxide vapor is delivered to one cylinder of a composite engine.

In heating the sulphur dioxide in the boiler, the air is reduced in temperature from 588 degrees above to 30 degrees below zero. It passes then to a "solair" reheater which raises it to 160 degrees. Then it goes into the air power cylinder of the engine and is worked by expansion down to atmospheric pressure and 60 degrees below zero. At this temperature it goes to a condenser where it reduces the exhaust from the sulphur dioxide cylinder into liquid of 40 degrees below zero in temperature. Leaving the condenser, the air passes to the atmosphere at practically atmospheric pressure and temperature.

The liquid sulphur dioxide is taken from the condenser by a pump and delivered to the boiler at 40 degrees below zero and 1000 pounds pressure. The air-compressing cylinder is jacketed with sulphur dioxide taken from the bottom of the boiler and brought back to a point high enough up to avoid excessive temperature differences.

By accepting the inventor's temperature ranges and calculating the heat interchanges and the work done in the two power cylinders and the compressing cylinder, it can be figured that the system is operative at an efficiency of at least 100 per cent. This, of course, is impossible, and it is an interesting problem to analyze the system and show where its fallacies are. We shall be glad to have our readers undertake this and advise us of their results.

Some Arithmetical Short Cuts

While the squaring of numbers is more frequently that of whole numbers than fractions, it is not infrequently necessary to square halves, quarters and eighths, such as 4½, 7½, etc., 5¼, 6¾, or 3⅛, 8⅛, or 9⅝, 2⅝, or any of these fractions.

Anyone who knows the multiplication table can square instantly any whole number up to 12. By remembering the following rule we can square any of the half numbers up to 12½ with equal facility.

Assume it to be desirable to multiply 7½ by 7½; simply multiply 7 by 8 and add ¼: thus $7 \times 8 = 56$, and $56 + ¼ = 56¼$, the answer. Or the value of $4½ \times 4½$ is found thus: $4 \times 5 = 20$, and $20 + ¼ = 20¼$. To do this requires no pencil or paper, and all that needs to be remembered is to add ¼. In like manner, though not so readily, the numbers with odd quarters may be squared; say, to find the value of $4¼ \times 4¼$, multiply 4 by 4½, which is 18, then add 1/16, which gives the right answer—18 1/16. This works out correctly whatever the odd quarter, and all that is necessary is to add 1/16, which, too, can be done without pencil or paper.

Similarly the odd eighths can be multiplied, but not so readily, except in the case of some numbers such as 7⅞ or 2⅛. Thus multiplying 7¾ by 8, we have $56 + (¾$ of 8 or) $6 = 62$, to which add 1/64, making 62 1/64, the answer. Or $2⅛ \times 2⅛$ is $2 \times 2¼ = 4½$, and 1/64 added, gives 4 33/64, the answer. This, too, can be done mentally, though odd numbers with odd eighths are more bothersome.

Anyone sufficiently interested can easily satisfy himself why the additional ¼, 1/16, etc., become necessary by drawing a diagram, Fig. 1, 2½ inches square, dividing it both ways into half inches, then taking a strip off the top and setting it up on end, Fig. 2, from which it will be seen that the result forms a rectangle 2x3 inches, and a half-inch square over, which is one-quarter of a square inch. This may not be new, but it is probably not well known, and it would seem to be worth any mechanic's time to learn, as are all short cuts to results.

There are short cuts that are not absolutely correct, and whether they should be used or not depends upon whether the result is as accurate as is required. The relation of the side of a square to its diagonal is approximately as 7 to 10, that is, it is within a small fraction; and whenever this ratio is used to determine a measurement on a drawing, such as the diagonal of a ⅞-inch square bolt head, and the resulting 1¼ inches is nearer than the draftsman can work, or needs to work, the short cut is good enough. So, too, the circumference of a circle is as 22 to 7 within 0.00001, which is far within the limits of a draftsman's ability to work, and probably far more accurate than can be obtained by the slide rule. These ratios, that is, 7 to 10 (or almost exactly 7 to 9.9) and 7 to 22, permanently drawn on a draftsman's drawing board (see Figs. 3 and 4), enable him to instantly determine the circumference of any circle within the range of the diagram or the diameter of any circle of which he has the circumference, or, from the other triangle, to determine the distance across the corners of any square within its range.

That is to say, if he wishes to know the distance across the corners of a 2¼-inch square he has only to measure off from the foot of the diagonal, as at *A*, Fig. 3, 2¼ inches, strike the vertical parallel to the original line *B*—*C* and the length of the new line is the distance required. In the case of the circumference of the circle, if the diameter is laid off from *A*, Fig. 4, down the vertical, as at *B*, and the horizontal parallel to the base is drawn through *B*, the length of the new line is the required circumference; or, if the circumference is known and the diameter is required, measure the circumference from *C* on the base line, as at *D*, strike the vertical and its length is the diameter. A second advantage in these diagrams is that, no matter what the measure or how fine the fraction, the result is obtained just as readily as with whole numbers, whereas, with the slide rule, everything is in tenths.

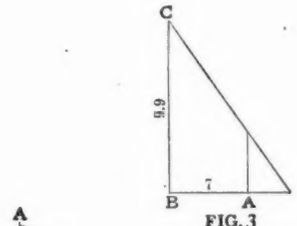


FIG. 3

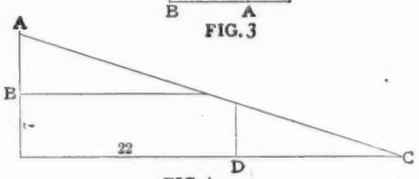


FIG. 4

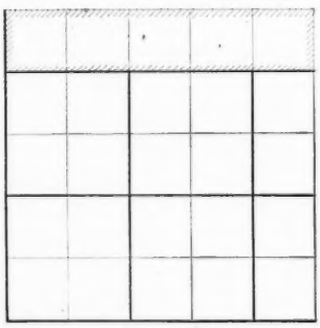


FIG. 1

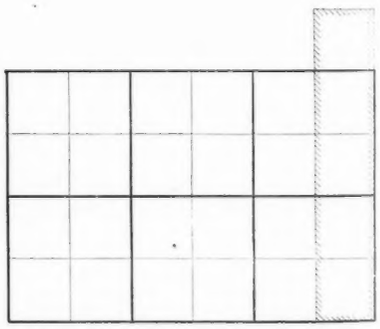


FIG. 2

PROOF OF THE RULE FOR SQUARES

When the measure wanted is small in proportion to the triangle, it is only necessary to use large units for the small ones needed. In making drawings to scale, say one-quarter size, we use small units to represent large ones, and in this case we can reverse the thing and use large units for small ones.—John E. Sweet in *American Machinist*.

Formulas for the Flow of Steam in Pipes

Comparison of Accepted Formulas, with Curves Showing the Drop in Pressure for Different Pipe Sizes and Velocities

BY PROF. G. F. GEBHARDT

The several accepted formulas relating to the flow of steam in pipes have been based upon a few experiments limited to pipes of small diameter, hence the application of these formulas to larger pipes, or to conditions other than those under which they were deduced, is apt to lead to considerable error. In small power plants extreme accuracy in determining the proper pipe sizes is not necessary; it is better to err in the installation of too large a pipe than one too small. In larger stations, however, when the pipes are large and the pressure high, the cost of piping increases very rapidly with the size. For example, the cost of 10-inch high-pressure fittings is from 15 to 20 per cent. greater than 9-inch fittings, and in large installations this first-cost item may be of considerable importance.

The practical problem ordinarily assumes the following form: Given the desired pressure at the throttle, the maximum weight of steam per unit of time, and the length of pipe, determine the minimum diameter of pipe which will conduct the given quantity of steam from the boiler to the engine without undue drop in pressure.

The simplest and most commonly used formula is based upon an allowable steam velocity of about 6000 feet per minute, friction and other causes of drop in pressure being disregarded; thus, for a velocity of 6000 feet per minute:

$$d = 0.175 \left(\frac{w}{y}\right)^{\frac{1}{2}} \qquad (1)$$

in which

d = diameter of the pipe in inches,
y = density of the steam in pounds per cubic foot,
and w = weight of steam flowing in pounds per minute.

In determining the diameter of the steam-pipe opening for reciprocating engines, a much lower velocity than 6000 feet per minute is assumed, to allow for the various conditions of operation. Average practice gives the constant in equation (1) a value of 0.3 instead of 0.175, when used in this connection.

Equation (1) gives satisfactory results for pipes under 100 feet in length, and between 4 and 8 inches in diameter; for larger diameters the velocity could be increased with advantage; for smaller diameters or greater lengths friction and condensation would cause considerable drop in pressure, and some one of the approved formulas analyzed further on should be used instead.

A large drop in pressure means a small pipe and high velocity, with consequent decrease in condensation, but a point is soon reached where the economy in the size of pipe is more than offset by the loss in friction. Some drop in pressure is necessary, otherwise there would be no flow; but there seems to be no fixed rule for determining the drop most suitable for any given set of conditions. In current practice the drop in pressure between boiler and engine ranges from a fraction of one pound to four pounds per square inch per 100 feet of pipe, with an average between one and two pounds.

TABLE I. FORMULAS FOR THE FLOW OF STEAM IN PIPES

No. of Equation.	Author.	Reference.	Formula As Given by the Author. *	Formula Reduced to Equivalent Notations. English Units. †
11	G. H. Babcock	"Steam" (1890). Pub. by B. & W. Co.	$W = 87 \left\{ \frac{y P d^5}{L\left(1 + \frac{3.6}{d}\right)} \right\}^{\frac{1}{2}}$	$W = 87 \left\{ \frac{y P d^5}{L\left(1 + \frac{3.6}{d}\right)} \right\}^{\frac{1}{2}}$
12	R. C. Carpenter	*Trans.* A. S. M. E., Vol. XX, p. 347.	$P = 0.0001306 \frac{W^2 L\left(1 + \frac{3.6}{d}\right)}{y d^5}$	$P = 0.0001306 \frac{W^2 L\left(1 + \frac{3.6}{d}\right)}{y d^5}$
7	Gutermuth	*Zeit. d. Ver. D. Ing.* Apr. 16, 1904, p. 562.	$P = 0.0015\, y \frac{L}{d} V^2$	$P = 105.8 \cdot 10^{-10}\, y \frac{L}{d} V^2$
9	Geipel and Kilgour	London *Electrician*, May 5, 1893.	$P = \frac{W^2 L}{2525\, y\, d^5}$	$P = \frac{W^2 L}{2525\, y\, d^5}$
5	Hawksley	*Pro.* Inst. C. E., Vol. XXXIII, p. 55.	$V = 48 \left\{ \frac{H d}{L} \right\}^{\frac{1}{2}}$	$V = 9976 \left\{ \frac{P d}{y L} \right\}^{\frac{1}{2}}$
15	Hurst	King's "Treatise on Coal Gas," Vol. II, p. 374.	$Q = 1350 \left\{ \frac{D^5 P}{G L} \right\}^{\frac{1}{2}}$	$V = 10{,}360 \left\{ \frac{P d}{y L} \right\}^{\frac{1}{2}}$
14	Ledoux	*Annales des Mines*, Vol II, 1892.	$d = 0.056 \left\{ \frac{W^2 L}{P_0^{1.94} - P_1^{1.94}} \right\}^{\frac{1}{5}}$	$d = 0.699 \left\{ \frac{W^2 L}{P_1^{1.94} - P_2^{1.94}} \right\}^{\frac{1}{5}}$
9a	Martin	*Engineering*, March 19, 1897.	$P = \frac{W^2 L}{3192\, y\, d^5}$	$P = \frac{W^2 L}{3192\, y\, d^5}$
4	Unwin	"Encyclopedia Britannica," Vol. XII, pp. 508, 516.	$H = 0.0027 \left(1 + \frac{3}{10\, D}\right) \frac{V^2}{2g} \cdot \frac{4L}{D}$	$P = 0.0001306 \frac{W^2 L\left(1 + \frac{3.6}{d}\right)}{y d^5}$

* See text for notations.

† All notations as follows:
P_1 = Initial pressure, lb. per sq. in., abs.
P_2 = Final pressure, lb. per sq. in., abs.
$P = P_1 - P_2$ = Drop in pressure.
W = Weight of steam flowing, lb. per min.
L = Length of pipe, ft.
d = Diameter of pipe, in.
y = Mean density, lb. per cu. ft.

COMPARISON OF FORMULAS

Considering the steady flow of steam in a pipe of uniform diameter, experiments show that the drop in pressure which takes place between any two points is influenced by the length and diameter of the pipe, the nature of the interior surface of the pipe, the velocity, density and quality of the steam, radiation, expansion and the effect of gravity. Attempts to include

these various factors in determining the proper size pipe have led to unwieldy formulas and have met with little encouragement. The numerous formulas employed in this connection are all based upon the hydraulic equation for the loss of head due to friction when water, steam or gas of any kind flows through a straight pipe, thus:

$$H = f \frac{v^2}{2g} \cdot \frac{4L}{d}, \qquad (2)$$

in which

H = loss of head measured in feet,
v = velocity of the fluid in feet per second,
L = length of the pipe in feet,
d = diameter of the pipe in feet,
g = acceleration of gravity = 32.2,
and f = coefficient of friction.

Data relative to the value of f are widely discordant. Eytelwein, Arson and Weisbach express it as a function of the velocity, and D'Arcy and Unwin as a function of the diameter. According to the former, the value of f is approximately constant for a given fluid, given character of surface, and small range of velocity. According to the latter:

$$f = K\left(1 + \frac{3}{10d}\right), \qquad (3)$$

in which

K = a factor determined by experiment,
and d = the diameter of the pipe in feet.

As will be shown later, *all of the most commonly used formulas for the flow of steam in pipes are but transpositions of equation (2)*, except in respect to the value of f, and may be divided into two main groups according to whether f is taken as a constant or a function of d.

The following values of K as applied to formula (3) have been given:

Unwin*, $K = 0.0027$ to 0.003.
Carpenter†, $K = 0.0018$ to 0.00276 (covered pipe).
Carpenter, $K = 0.00098$ to 0.0036 (uncovered pipe).
Arson†, $K = 0.005$.
Arson, $K = 0.0028$ (for flow of air by experiment at St. Gothard's tunnel).‡

The average of these values is not far from 0.0027, the figure ordinarily used.

By combining equations (2) and (3) and substituting for f the value 0.0027, the formula given by Professor Unwin is obtained:

UNWIN'S FORMULA*:

$$H = 0.0027\left(1 + \frac{3}{10d}\right)\frac{4L}{d} \cdot \frac{V^2}{2g}. \qquad (4)$$

By giving f the value 0.007 and solving for V, equation (2) assumes the form given by Hawksley:

HAWKSLEY'S FORMULA||:

$$V = 48\sqrt{\frac{Hd}{L}}, \qquad (5)$$

which is but a modification of D'Arcy's well-known formula for the flow of water:

$$V = c\sqrt{\frac{Hd}{4L}}.$$

The constant 48 refers properly to water only, though it is often used in connection with the flow of steam.

Substituting the value $\frac{p}{y}$ for H in equation (2) and solving for p, we obtain:

$$p = A\frac{yL}{d}V^2, \qquad (6)$$

in which A is a factor including all the constants of conversion. If A is given the value 0.0015, the equation takes the form given by Gutermuth (*Zeit. d. Ver. D. Ing.*, 1887, p. 718):

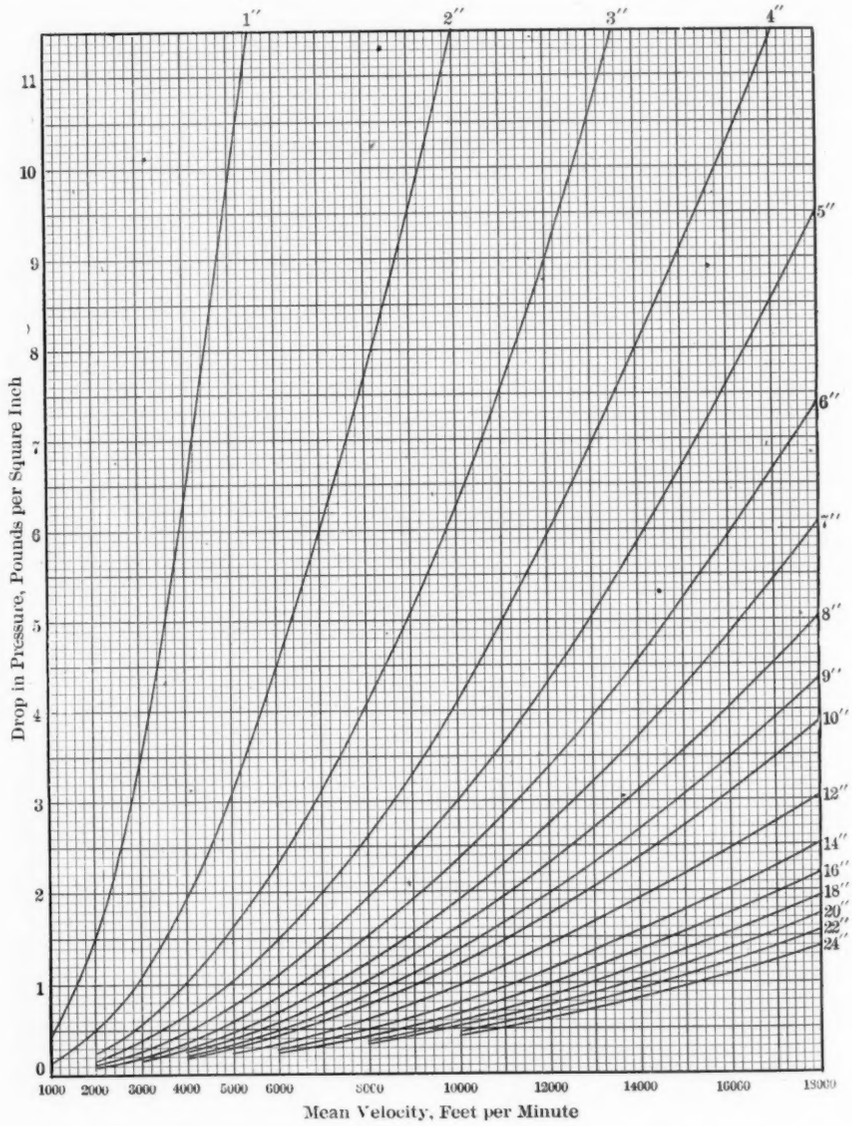
GUTERMUTH'S FORMULA:

$$p = 0.0015\, y \frac{L}{d} V^2, \qquad (7)$$

in which p is in kilograms per square meter, d and L in meters, V in meters per second, and y in kilograms per cubic meter.

Substitute for V in equation (2) its value, $\frac{4w}{y\pi d^2}$, and for H its value, $\frac{p}{y}$, and solve for p:

$$p = B\frac{w^2 L}{y d^5}, \qquad (8)$$



in which B is a factor including the various constants of conversion.

If B has the value $\frac{1}{2525}$, the equation assumes the form proposed by Geipel and Kilgour (London *Electrician*, May 5, 1893):

GEIPEL AND KILGOUR'S FORMULA:

$$p = \frac{w^2 L}{2525\, y\, d^5}, \qquad (9)$$

in which p is in pounds per square inch, w in pounds per minute, L and d in inches and y in pounds per cubic foot. A. J. Mar-

*Encyclopedia Britannica, Vol. XII, pp. 508, 516.
†*Trans.* A. S. M. E., Vol. XX, p. 355.
‡Van Nostrand's *Engineering Magazine*, Vol. 24, p. 96.
||*Pro.* Inst. C. E., Vol. XXXIII, p. 55.

tin (*Engineering*, March 19, 1897, p. 361), gives B a value, 1/3192.

Combining equations (2) and (3), and substituting $\frac{4\,w}{y\,\pi\,d^2}$ for V and $\frac{p}{y}$ for H, we obtain the expression:

$$w = C\sqrt{\frac{y\,d^5\,p}{L\left(1+\frac{3}{10\,d}\right)}}, \qquad (10)$$

in which C is a factor incuding all the constants of conversion.

If C is made equal to 87, the equation assumes the form proposed by G. H. Babcock ("Steam," published by Babcock & Wilcox Company):

BABCOCK'S FORMULA:

$$w = 87\sqrt{\frac{y\,d^5\,p}{L\left(1+\frac{3.6}{d}\right)}}, \qquad (11)$$

in which

w = the weight of steam in pounds, per minute,
d = the diameter of the pipe in inches,
L = the length of pipe in feet,
p = the drop in pressure in pounds per square inch,
and y = the *mean* density in pounds per cubic foot.

length. The equation as used by him takes the following form:

CARPENTER'S FORMULA:

$$P = 0.000131\,\frac{w^2\,L\left(1+\frac{3.6}{d}\right)}{y\,d^5}. \qquad (12)$$

M. Ledoux conducted experiments on pipes 47, 71, 75, and 100 millimeters in diameter (1.85, 2.79, 2.95, 3.94 inches) and with lengths varying from 100 to 330 meters (328 to 1082 feet). Reduced to the form of equation (11), the Ledoux formula gives the constant an average value of 87.45. (*Trans.* A. S. M. E., Vol. XX, p. 365.)

Ledoux's formula is obtained from equation (8), which, expressed in terms of d, becomes:

$$d = F\sqrt[5]{\frac{w^2\,L}{p\,y}}, \qquad (13)$$

F being the factor of conversion. If F has the value of 0.056 and y is expressed in terms of an exponential function of p, the equation becomes:

LEDOUX'S FORMULA:

$$d = 0.056\sqrt[5]{\frac{w^2\,L}{p_1^{\,1.94}-p_2^{\,1.94}}}, \qquad (14)$$

in which d and L are in meters, w in kilograms per second, and $p = p_1 - p_2$ in atmospheres.

where

Q = cubic feet of gas per hour,
D = diameter of pipe in inches,
L = length of pipe in yards,
p = pressure in inches of water,
and G = specific gravity of gas, air unity.

Molesworth uses this equation for the flow of steam, giving the constant a value of 1000.

Since all of the above formulas are used more or less by engineers, in determining the proper proportions of pipes for the conveyance of steam, and yet, in some cases give widely discordant results, the desire naturally arises to learn something about their relative merits. In order to facilitate comparison, Table 1 has been constructed. Here the several equations are tabulated as originally proposed, together with the author's name, and with reference to the article wherein they were first published. The equivalent of each formula is also given expressed in the units ordinarily used in this country, in order to simplify comparison by numerical substitution.

It should be noted that while the different forms bear little or no obvious resemblance to each other, and are quite complex in appearance, they are, never-

TABLE 2. COMPARISON OF PIPE FORMULAS FOR THE FLOW OF STEAM

	Author.	V = Velocity, Feet per minute.	W = Weight, Lb. per minute.	P = Drop in Pressure, Lb. per sq. inch.	d = Diameter, Inches.
GROUP II.	Geipel and Kilgour.	$V = 9240\sqrt{\frac{P\,d}{y\,L}}$	$W = 50.2\sqrt{\frac{P\,y\,d^5}{L}}$	$P = 0.0003960\,\frac{W^2 L}{y\,d^5}$	$d = 0.2087\sqrt[5]{\frac{W^2\,L}{P\,y}}$
	Gutermuth...	$V = 9722\sqrt{\frac{P\,d}{y\,L}}$	$W = 53\sqrt{\frac{P\,y\,d^5}{L}}$	$P = 0.0003557\,\frac{W^2\,L}{y\,d^5}$	$d = 0.2032\sqrt[5]{\frac{W^2\,L}{P\,y}}$
	Hawksley....	$V = 9976\sqrt{\frac{P\,d}{y\,L}}$	$W = 54.4\sqrt{\frac{P\,y\,d^5}{L}}$	$P = 0.0003370\,\frac{W^2\,L}{y\,d^5}$	$d = 0.2010\sqrt[5]{\frac{W^2\,L}{P\,y}}$
	Martin.....	$V = 10{,}350\sqrt{\frac{P\,d}{y\,L}}$	$W = 56.5\sqrt{\frac{P\,y\,d^5}{L}}$	$P = 0.0003133\,\frac{W^2\,L}{y\,d^5}$	$d = 0.1990\sqrt[5]{\frac{W^2\,L}{P\,y}}$
	Hurst.....	$V = 10{,}360\sqrt{\frac{P\,d}{y\,L}}$	$W = 56.5\sqrt{\frac{P\,y\,d^5}{L}}$	$P = 0.0003126\,\frac{W^2\,L}{y\,d^5}$	$d = 0.1990\sqrt[5]{\frac{W^2\,L}{P\,y}}$
GROUP I.	Babcock	$V = 15{,}950\sqrt{\frac{P\,d}{y\,L\left(1+\frac{3.6}{d}\right)}}$	$W = 87\sqrt{\frac{P\,y\,d^5}{L\left(1+\frac{3.6}{d}\right)}}$	$P = 0.0001321\,\frac{W^2\,L\left(1+\frac{3.6}{d}\right)}{y\,d^5}$	.....
	Unwin	$V = 16{,}050\sqrt{\frac{P\,d}{y\,L\left(1+\frac{3.6}{d}\right)}}$	$W = 87.5\sqrt{\frac{P\,y\,d^5}{L\left(1+\frac{3.6}{d}\right)}}$	$P = 0.0001306\,\frac{W^2\,L\left(1+\frac{3.6}{d}\right)}{y\,d^5}$	.....
	Carpenter....	$V = 16{,}050\sqrt{\frac{P\,d}{y\,L\left(1+\frac{3.6}{d}\right)}}$	$W = 87.5\sqrt{\frac{P\,y\,d^5}{L\left(1+\frac{3.6}{d}\right)}}$	$P = 0.0001306\,\frac{W^2\,L\left(1+\frac{3.6}{d}\right)}{y\,d^5}$	.....
	Ledoux.....	$V = 442\sqrt{\frac{(P_1^{\,1.94}-P_2^{\,1.94})\,d}{y^2\,L}}$	$W = 2.44\sqrt{\frac{(P_1^{\,1.94}-P_2^{\,1.94})\,d^5}{L}}$	$P_1^{\,1.94}-P_2^{\,1.94} = 0.1669\,\frac{W^2\,L}{d^5}$	$d = 0.699\sqrt[5]{\frac{W^2\,L}{P_1^{\,1.94}-P_2^{\,1.94}}}$

It should be noted that the above is Unwin's equation (4) expressed in terms of p and w, and is its exact equivalent in all respects. This is probably the most widely accepted formula for the flow of steam and agrees very closely indeed with the results of experiments conducted by Prof. R. C. Carpenter (*Trans.* A. S. M. E., Vol XX, p. 355), and by M. Ledoux (*Annales des Mines*, Vol. 11, 1892). Professor Carpenter determined the value of the constant from experiments conducted on pipes 1 inch, 1½, 2, 3 and 5 inches in diameter, and from 90 to 250 feet in

J. T. Hurst (see King's "Treatise on Coal Gas," Vol. II, p. 374) gives the following formula based on experiments conducted with the flow of coal gas in pipes, and although not primarily intended for use with steam is frequently used in this connection:

HURST'S FORMULA:

$$Q = 1350\sqrt{\frac{D^2\,p}{G\,L}}, \qquad (15)$$

theless, *merely transpositions of the fundamental equation* (2), except in respect to the value of the coefficient of friction f. The formulas proposed by Babcock, Unwin and Carpenter are based on a value of

$$f = K\left(1+\frac{3}{10\,d}\right),$$

and those proposed by Gutermuth, Geipel and Kilgour, Hawksley, Hurst and Martin are based on a constant value for f. This divides them into two groups. Within each group the range in results is not

material, but the results of the groups differ widely. The formulas in the former group are based on comparatively recent experiments with steam, whereas the latter are deduced, for the most part, from experiments with compressed air and water; hence, the former should be the more reliable. Ledoux's formula stands in a class of its own, but as it agrees more closely with the formulas in group 1 than those in group 2, it is classified with the former.

Table 3 illustrates an application of the formulas to a number of concrete examples and shows the drop in pressure for different diameters of pipes and for a large range in steam velocities. In this table the grouping of the formulas is made clear. The results in the first group, with the exception of those deduced from Ledoux's formula, are practically identical, and those in the second group differ but slightly from each other; but the two groups differ considerably.

The formulas which have been cited are strictly applicable only to well lagged pipes free from bends, or obstructions of any kind, such as valves or fittings, all of which greatly increase the frictional resistance to the flow of steam. If these obstructions must be considered, it is customary to allow for them by assuming an added length of straight pipes, equivalent in resistance to the various fittings and bends. The few tests which have been made for the purpose of determining the resistance of various pipe fittings unfortunately give very discordant results, and in the absence of more recent data the rules given by Briggs* are probably as accurate as any.

According to Briggs, the length of pipe in inches equivalent to one standard 90-degree elbow is:

$$L^1 = 76\,d \div \left(1 + \frac{3.6}{d}\right), \quad (16)$$

and to one globe valve:

$$L^1 = 114\,d \div \left(1 + \frac{3.6}{d}\right). \quad (17)$$

The resistance of gate valves is negligible.

*Briggs' "Warming Buildings by Steam."

The drop in pressure in actual installations is much greater than is generally supposed. Table 4, taken from the valuable paper of P. Berner‡ on the "Flow of Superheated Steam," gives some idea of the actual drop in pressure and temperature, and the condensation in German power plants working under normal conditions for both saturated and superheated steam. The units in this table have been converted to the British system to facilitate comparison. Unfortunately, the number and character of fittings are not stated, and hence no general comparison can be made between the actual results and those computed from the formulas.

In the absence of data to the contrary, formulas 4, 11, 12 and 14 may be taken as representative of current practice in steam power-plant work for both superheated and saturated steam.

The curves in the accompanying diagram have been calculated by the author with the average of the formulas in group 1, and are of interest in showing the drop in pressure for different pipe sizes and steam velocities. The curves are based on a steam pressure of 100 pounds absolute and pipe lengths of 100 feet. Within the limit of 12,000 feet per minute velocity and 10 pounds per square inch drop in pressure, the curves are sufficiently accurate for all practical purposes, but beyond this range the results are purely conjectural, and may or may not be accurate, as no recorded experin have been

‡*Zeit d. Ver. D. Ing.*, April 16, 1904.

TABLE 4. TESTS OF THE FLOW OF SATURATED VS. SUPERHEATED STEAM IN ACTUAL POWER PLANTS UNDER NORMAL CONDITIONS

No. of Station	Condition of Steam.	d	L	P_1	t_1	P_2	t_2	P	W	V	w
1	Saturated	2.56	492	82	313.5	78	308.3	1.21	42.4	6500	0.205
	Superheated			82.7	772	74	464	1.43	23.6	5220	0.328
2	Saturated	4.73	82	146.6	356.4	143.9	355	3.46	75.3	1910	0.676
	Superheated		79	150.2	582	147.1	542	4.16	53.2	1682	0.168
3	Saturated	4.92	177	154 6	360.5	151	358.7	2.17	246	5450	6.121
	Superheated			145.5	410	133.8	370	6.6	351.8	8980	0.037
4	Saturated	5.92	98.5	182.1	373	170.9	368.6	11.5	158	2420	0.143
	Superheated			183.8	464.5	173.6	457	10.2	145	2181	0.074
5	Saturated	5.92	210	123.1	342.9	115.2	337.9	3.72	75.4	1475	0.158
	Superheated			121.9	556	115.8	409	2.86	68.5	1555	0.037
6	Saturated	7.87	128	121	341.6	111.8	355.7	7.67	48	531	.....
	Superheated			121.8	514	114.8	390	5 34	42.3	551	.....
7	Saturated	8.28	192	108.8	333.8	107.6	332.9	.....	146	1592	0.536
	Superheated			108.9	547	108	470	.....	119.5	1675	0.177
8	Superheated	8.87	86	246	476	189.1	439	16.4	730	4320	0.639
9	Saturated	8 87	181	128	345.9	123.9	343.5	2.38	68.5	492	0.822
	Superheated			128	459	123.9	350	2.38	52.7	473	0.252
10	Saturated	9.07	131	136.9	351.1	131	347.6	4.55	122.4	905	0.094
	Superheated			135.1	470	131	416	3.12	103	778	0.067
11	Saturated	9.47	95	104.4	330.7	97.6	325.8	7.2	238	2490	0.200
	Superheated			111	490.2	105.2	456	5.98	193	2022	0.020
12	Saturated	9.85	126	170.9	368.6	165.1	365.7	4.5	163	827	0.702
	Superheated			172.2	477	169.3	426	2.25	143	807	0.096

d = Diameter of pipe in inches.
L = Length of pipe in feet.
P_1 = Initial pressure, lb. per sq. in., abs.
P_2 = Final pressure, lb. per sq. in., abs.
V = Mean velocity, feet per minute.
t_1 = Initial temperature, degrees F.
t_2 = Final temperature, degrees F.
P = Drop in pressure per 100 feet of pipe, including resistance of superheater.
W = Weight of steam, pounds per min.
w = Pipe condensation, lb. per sq. ft. per hr.

TABLE 3. COMPARISON OF FORMULAS FOR THE FLOW OF STEAM IN PIPES

Drop in pressure, pounds per square inch.

$L = 100$ feet. $P_1 = 100$ lb. per sq. in. absolute.

	Velocity, Feet per minute.	2000					4000					8000				12,000			
	Diameter, Inches.	2	4	8	12	16	2	4	8	12	16	4	8	12	16	4	8	12	16
GROUP I.	Babcock, Carpenter, Unwin	0.49	0.16	0.06	0.04	0.03	1.96	0.65	0.25	0.15	0.11	2.63	1.00	0.62	0.44	5.86	2.26	1.35	0.98
	Ledoux	0.35	0.15	0.06	0.04	0.03	1.35	0.7	0.36	0.23	0.17	2.6	1.35	0.95	0.65	6.0	3.04	2.00	1.53
GROUP II.	Geipel and Kilgour	0.51	0.25	0.13	0.09	0.07	2.14	1.07	0.57	0.35	0.26	4.25	2.15	1.47	1.08	9.32	4.72	3.16	2.58
	Gutermuth	0.48	0.24	0.12	0.08	0.06	1.92	0.96	0.48	0.32	0.24	3.84	1.92	1.29	0.97	8.43	4.23	2.86	2.16
	Hawksley	0.45	0.22	0.11	0.08	0.06	1.82	0.91	0.45	0.30	0.22	3.58	1.81	1.21	0.91	7.9	4.04	2.72	2.04
	Hurst, Martin	0.42	0.21	0.10	0.07	0.05	1.69	0.84	0.42	0.28	0.21	3.34	1.70	1.13	0.85	7.4	3.74	2.5	1.89